A TEXT-BOOK

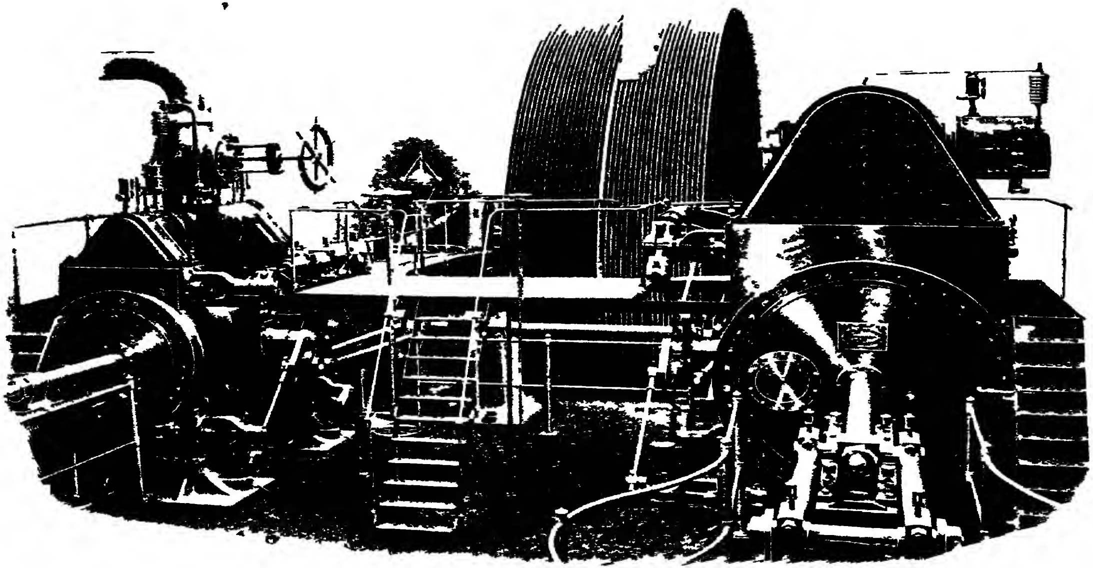
ON

STEAM AND STEAM ENGINES,

INCLUDING

TURBINES AND BOILERS.

FIFTEENTH EDITION.



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COMPOUND COUPLED CONDENSING CORLISS VALVE ENGINE 2500 I H P

H P CYLR 34 DIAR L P CYLR 70 DIAR EACH 6 FEET STROKE
FLYWHEEL 110 TONS 30 FEET DIAR WITH 38 ROPES OF 1 DIAR
STEAM PRESSURE 160 POUNDS PER SQUARE INCH VACUUM 26 INCHES 60 REVS PER MIN

See Prof A Jameson's Text Book on Steam, &c for Corliss Engines

A TEXT-BOOK

OF

STEAM AND STEAM ENGINES,

INCLUDING

TURBINES AND BOILERS.

Specially Arranged
For the Use of Engineers Qualifying for the Institution of Civil Engineers, The Diplomas and Degrees of Technical Colleges and Universities, Advanced Science Certificates of British and Colonial Boards of Education, and Honours Certificates of the City and Guilds of London Institute, in Mechanical Engineering, and Engineers Generally.

BY

ANDREW JAMIESON, M.INST.C.E.,

FORMERLY PROFESSOR OF ENGINEERING IN THE GLASGOW AND WEST OF SCOTLAND TECHNICAL COLLEGE; MEMBER OF THE INSTITUTION OF ELECTRICAL ENGINEERS; FELLOW OF THE ROYAL SOCIETY, EDINBURGH; AUTHOR OF TEXT-BOOKS ON STEAM AND STEAM ENGINES, TURBINES, APPLIED MECHANICS, MAGNETISM AND ELECTRICITY, ELECTRICAL RULES AND TABLES, ETC.

FIFTEENTH EDITION, REVISED.

With Numerous Diagrams, Folding-Plates, and Examination Questions.

LONDON:
CHARLES GRIFFIN & COMPANY, LIMITED;
EXETER STREET, STRAND.
1906.

PREFACE

TO THE FIFTEENTH EDITION.

THE whole of the Fourteenth Edition has been carefully revised, whilst the questions at the end of each lecture have been worked out by the author's "Engineering Correspondence Students" and duly checked.

The 1905 Examination Papers of The Institution of Civil Engineers on "The Theory of Heat Engines," Stages II. and III. of the 1905 Board of Education Papers on "Steam," as well as the 1905 Mechanical Engineering Questions of The City and Guilds of London Institute relating to "The Theory of Steam Engines and the Thermodynamics of Steam" have been included under Appendix D.

In Appendix E, Lectures XIII., XVI., and XX. have been extended to include a special plate and description of the new "Cipollina" duplex indicator, whereby simultaneous diagrams are obtained from each end of a cylinder; a complete, detailed, illustrated description of the most economical steam engine on record; plates of Corliss engines; a table of some of the best and most reliable results of non-condensing, compound, triple-expansion engines, and recent steam turbine trials. These additions now bring this book up to a total of 816 text pages, with 26 preliminary pages.

I have much pleasure in thanking those firms who kindly gave me drawings and data, my engineering correspondence students for hints, and, more especially, Mr. John Ramsay, Assoc.M.Inst.C.E., for his help in preparing this Fifteenth Edition for the press.

ANDREW JAMIESON.

16 ROSSLYN TERRACE, KELVINSIDE,
GLASGOW, *January*, 1906.

The most economical quadruple-expansion marine engines and their boilers, together with their appliances for induced draught, superheating, jacketing, and draining—which led to their unsurpassed economy of less than one pound of coal per I.H.P.-hour—have been very fully illustrated.

The latest forms of boilers, engines, and condensing plants for up-to-date Electric Light and Power Installations have been depicted and described.

The causes of, and remedies for, corrosion in marine condensers and boilers have been discussed.

Two new Lectures (XXVI. and XXVII.) have been specially written upon the latest and best Steam Turbines for land and marine purposes, with a mathematical explanation of their steam consumption. These lectures, together with the latest available information on superheaters, will, it is hoped, prepare engineering students for the rapidly-coming change in the propulsion of vessels and the driving of Electric Power Plants.

At the end of each lecture there has been given a number of Questions in the order of treatment, as well as selected Examples from the Board of Education, City and Guilds of London Institute, and other sources, to enable teachers to test with a minimum of trouble how far their students understand the subject. Further, every question which has yet been set by The Institution of Civil Engineers on "The Theory of Heat Engines" for their Associate Membership will be found in Appendix A. These questions have been arranged in the order of their dates under the respective lecture numbers.

The latest (1904) Board of Education "Steam," and the City and Guilds of London Mechanical Engineering Questions on "Steam Engines" will be found in Appendices B and C.

In Appendix D extracts from the present 1904 Rules and Syllabus of the Institution of Civil Engineers are printed as a guide to intending candidates for admission as Associate Members. Also, the questions set at the Feb., 1904, Examinations upon "The Theory of Heat Engines" have been given as an example of these well-conducted "Examinations for the Profession."

Each source of information to which the author is indebted has been duly acknowledged where it occurs. Lists of the best and latest papers read before The Institutions of Civil and Mechanical Engineers, &c., and many of the latest books upon the different sections have been given by footnotes, so that students may have little difficulty in extending their knowledge beyond the scope of this book.

I have much pleasure in thanking the several engineers and firms who kindly gave me information, drawings, and special blocks or plates. I have also to thank my chief assistant, Mr. John Ramsay, for his help in the preparation of this new Fourteenth Edition.

ANDREW JAMIESON.

16 Rosslyn Terrace, Kelvinside,
Glasgow, *October*, 1904.

ABSTRACT

OF

PREFACE TO THE FIRST EDITION.

In a leading article on educational Engineering Treatises, which appeared lately in a well-known journal, the following remarks, amongst others, struck me as being very suggestive to any one engaged in the preparation of a Text-Book for Students, and as well worthy of attention :—

"We are convinced that all the instruction contained in a great number of the engineering books already published, could be printed much more simply and concisely, and also much more lucidly, if authors sought only to impart their knowledge with the greatest brevity, without thinking at all of displaying their own learning or seeking to make a thick volume. . . . There is too much paste and scissors work, too much book-making and padding nowadays. . . . A considerable number of engineering books are so learned as to be quite over the heads of most students. Many more are so verbose, so laden with abstruse formulæ, letters, and diagrams, that the solution of the simplest question involves hours of time that can ill be spared from other work. It is no doubt true, that many engineering questions demand elaborate writing to give a precise answer with mathematical exactness; but in the majority of engineering practice, absolute exactness of such a nature is not necessary, and if a useful approximation will amply suffice, and is readily obtainable in some simply written book, that is the one that will be adopted."

The object, therefore, aimed at in the following pages, was the production of such a "simply written book" as should *not* be above the heads of my readers, but should bring the information desired, step by step, within their grasp. Whether I have succeeded in accomplishing this object, is a question which, of course, must be decided by those competent to judge.

It is designed to be an easy introduction to Professor Rankine's well-known treatise on *The Steam Engine*, and to Mr. Seaton's practical and highly appreciated *Manual of Marine Engineering*, both issued by the publishers of the present volume.

ANDREW JAMIESON.

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MECHANICAL ENGINEERING SYMBOLS, ABBREVIATIONS AND INDEX LETTERS

USED IN PROFESSOR JAMIESON'S "APPLIED MECHANICS AND MECHANICAL ENGINEERING" AND "STEAM" BOOKS.

Prefatory Note.—It is very tantalising, as well as a great inconvenience to Students and Engineers, to find so many different symbol letters being used for denoting one and the same thing by various writers on mechanics. It is a pity, that British Civil and Mechanical Engineers have not as yet *standardised* their symbols in the same way that Chemists and Electrical Engineers have done. The Committee on Notation of the Chamber of Delegates to the International Electrical Congress, which met at Chicago in 1893, recommended a set of "Symbols for Physical Quantities and Abbreviations for Units," which have ever since been (almost) universally adopted throughout the world by Electricians.* This at once enables the results of certain new or corroborative investigations and formulæ, which may have been made and printed anywhere, to be clearly understood anywhere else, without having to specially interpret the precise meaning of each symbol letter.

In the following list of symbols, abbreviations and index letters, the *first* letter of the chief noun or most important word has been used to indicate the same. Where it appeared necessary, the *first* letter or letters of the adjectival substantive or qualifying words have been added, either as a following or as a subscript or suffix letter or letters. For certain specific quantities, ratios, coefficients and angles, small Greek letters have been used, and I have added to this list the complete Greek alphabet, since it may be refreshing to the memory of some to again see and read the names of these letters, which were no doubt quite familiar to them when at school.

* These "Symbols for Physical Quantities and Abbreviations for Units" will be found printed *in full* in the form of a table at the commencement of Munro and Jamieson's *Pocket-Book of Electrical Rules and Tables*. If a similar recommendation were authorised by a committee composed of delegates from the chief Engineering Institutions, it would be gladly adopted by "The Profession" in the same way that the present work of "The Engineering Standards Committee" is being accepted.

TABLE OF MECHANICAL ENGINEERING QUANTITIES, SYMBOLS, UNITS AND THEIR ABBREVIATIONS.

(As used in Prof. Jamieson's "Applied Mechanics" and "Steam" Books.)

Quantities.	Symbols.	Defining Equations.	Practical Units.	Abbreviations of the Practical Units.
FUNDAMENTAL.				
Length, . . .	L, l	...	Yard, ... Foot, ... Inch, ...	yd. ft. in.
Mass, . . .	M, m	...	Pound, . . .	lb.
Time, . . .	T, t	...	Second, . . . Minute, . . . Hour, . . .	s. m. h.
GEOMETRIC.				
Surface, . . .	S, s	$S = L^2$	Square foot, . . Square inch, . .	sq. ft. sq. in.
Volume, . . .	V	$V = L^3$	Cubic foot, . . Cubic inch, . .	cb. ft. cb. in.
Angle, ∠ . .	α, β θ, ϕ	$\alpha = \frac{\text{arc}}{\text{radius}}$	Degree, . . . Minute, . . . Second, . . . Radian $= \frac{180^\circ}{\pi}$.	1° $1'$ $1''$ rn.
MECHANICAL.				
Velocity, . .	v	$v = \frac{L}{T}$	Foot per second, .	$\frac{\text{ft.}}{\text{s.}}$
Angular velocity, .	ω	$\omega = \frac{v}{L} = \frac{\theta}{t}$	Revs. per second, . Revs. per minute, Radians per second,	r.p.s. r.p.m. ω
Acceleration, . .	a, g	$a = \frac{v}{T}$	Foot per sec. per sec.	$\frac{\text{ft.}}{\text{s}^2}$
Force, . . .	F, f	...	Pound weight (gravitational unit),	lb. wt. (or lb.)
	W, w	$F = Ma$	Poundal (absolute unit),	pdl.
Pressure (per unit area),	p	$p = \frac{F}{s}$	Pound per sq. inch,	lb. □"
Work, . . .	(Wh)	$Wh = FL$	Foot-pound, . .	ft.-lb.
Potential energy, .	E_P	$E_P = Wh$	Foot-pound, . .	ft.-lb.
Kinetic energy, .	E_K	$E_K = \frac{Wv^2}{2g}$	Foot-pound, . .	ft.-lb.
Power or activity, .	HP	$\text{H.P.} = \frac{Wh}{T}$	Horse power, . Ft.-lb. per min., . Ft.-lb. per sec., .	H.P. ft.-lb./m. ft.-lb./s.
Moment of inertia, .	I	$I = Mk^2$	... ...	lb.-ft.^{2}
Density, . . .	ρ	$\rho = \frac{M}{V}$	Pound per cb. ft., . Pound per cb. in., .	$\frac{\text{lb.}}{\text{ft.}^3}$ $\frac{\text{lb.}}{\text{in.}^3}$

OTHER SYMBOLS AND ABBREVIATIONS IN VOLS. I. AND II.

A for Areas.
B, b ,, Breadths.
C, c, k ,, Constants, ratios.
c.g. ,, Centre of gravity.
D, d ,, Diameters depths, deflections.
D_1, D_2, D_3 ,, Drivers in gearing.
E ,, Modulus of elasticity.
e ,, Velocity ratio in wheel gearing.
F_1, F_2, F_3 ,, Followers in gearing.
f_s, f_t ,, Forces of shear and tension.
H, h ,, Heights, heads.
H.P., h.p. ,, Horse-power.
B.H.P. ,, Brake horse-power.
E.H.P. ,, Effective ,,
I.H.P. ,, Indicated ,,
k ,, Radius of gyration, or, Coef. of discharge in hydraulics.
N, n ,, Numbers—*e.g.*, number of revs. per min., number of teeth, &c.
P, Q ,, Push or pull forces.
R_1 R_2 ,, Reactions, resultants, radii, resistances.
s ,, Seconds, space, surface. Displacement, distance.
S F ,, Shearing force.
T M ,, Torsional moment.
T R ,, Torsional resistance.
B M ,, Bending moment.
M R ,, Moment of resistance.
R M ,, Resisting moment.
T_d, T_s ,, Tensions on driving and slack sides of belts or ropes, &c.
W_L, W_T, W_U ,, Lost, total, and useful work.

x, y, z for Unknown quantities.
Z ,, Modulus of section.
Z_t ,, ,, tension.
Z_c ,, ,, compression.

δ, d for Differential signs which are prefixed to another letter; then the two together represent a very small quantity.
ϵ, e ,, Represents base of Naperian Logs = 2·7182; for example, $\log_e 3 = 1{\cdot}1$.
η ,, Efficiency.
λ ,, Length ratio of ship to model.
μ ,, Coefficient of friction.
π ,, Circumference of a circle ÷ its diameter.
ρ ,, Radius of curvature, radian.

Σ for Symbol for sum total of a number of quantities.
$\int_0^x$,, Sign of integration or summation between limits 0 and x.
$\sim$,, Sign for the difference between two quantities.
$\square$,, Sign for square—*e.g.*, 10 $\square''$ = 10 square inches.
— ,, Sign over two letters, $\overline{PQ}$, for a force acting from P to → Q, means that they represent a vector quantity, which has (1) magnitude, (2) direction, (3) sense.
$\geqq$,, Sign for equal to or greater than.
$\leqq$,, Sign for equal to or less than.

GREEK ALPHABET.

A	α	Alpha.	I	ι	Iota.	P	ρ	Rho.
B	β	Beta.	K	κ	Kappa.	Σ	σ or ς	Sigma.
Γ	γ	Gamma.	Λ	λ	Lambda	T	τ	Tau.
Δ	δ	Delta.	M	μ	Mu	Υ	υ	Upsilon.
E	ϵ	Epsilon.	N	ν	Nu	Φ	ϕ	Phi.
Z	ζ	Zeta.	Ξ	ξ	Xi.	X	χ	Chi.
H	η	Eta.	O	o	Omicron	Ψ	ψ	Psi.
Θ	θ	Theta.	Π	π	Pi.	Ω	ω	Omĕga.

ADDITIONAL SYMBOLS AND ABBREVIATIONS USED IN THIS STEAM BOOK.

B.T.U. for British thermal units.
c ,, Clearance volume.
C_V ,, Calorific value.
E, E_f ,, Evaporation factor.
E_I ,, Internal energy.
h ,, heat any (sensible) small quantity.
H, H_T ,, Total heat of evaporation.
H_E ,, Heat (external) or work done during evaporation.
H_I ,, Heat (internal) or work done during evaporation.
H_{sa} ,, Heat units per lb. of saturated steam
H_{su} ,, Heat units per lb. of superheated steam.
H_σ ,, Heat (specific) of steam.
J ,, Joule's mechanical equivalent of the unit of heat.
L ,, Latent heat of steam.
p ,, Pitch of rivets in joints.
p_b ,, Pressure (back) in lbs. per sq. in.
p_m ,, Pressure (mean) in lbs. per sq. in.
P ,, Pressure (total).
P_S ,, Pitch of a screw propeller.
Q ,, Quantity of heat.
dQ ,, Minute quantity of heat.
r for Ratio of expansion; radius of crank-pin circle.
r.p.m. ,, Revolutions per minute.
S, s ,, Sensible heat; Slip of screw propeller; Speed of piston.
S_s ,, Shearing strength of rivets per sq. in.
S_t ,, Tensile strength of plates per sq. in.
t ,, Thickness.
t_f ,, Temperature of feed.
t_{su} ,, Superheat at steam chest in degrees Fah.
$t°$, $t_1°$,, Temperature in degrees.
τ ,, Absolute temperatures.
T ,, Travel of slide valve.
T_S ,, Thrust of a screw propeller.
u ,, Units of heat as a suffix—*e.g.*, $10\,u$ = ten units.
V ,, Velocity.
V_s ,, Volume of 1 lb. of dry steam.
V_w ,, Volume of 1 lb. of water.
V_{ws} ,, Volume of 1 lb. of wet steam.
W ,, Work done, load or weight.
W_{sa} ,, Weight of saturated steam.
W_u ,, Weight of superheated steam.
λ, ω ,, Angles, $\angle$.

APPLIED MECHANICS AND MECHANICAL ENGINEERING.

VOLUME I.

Of 594 *Pages;* 300 *Figures and* 540 *Questions.*

PART I.—The Principle of Work and its Applications; Friction, Power Tests, with Efficiencies of Machines.

PART II.—Tooth, Friction, Belt, Rope, Chain and Miscellaneous Gearing, with their Applications to Machines.—Shapes and Strengths of Teeth.—Automatic Tooth-Cutting Machines.—Velocity-Ratio and Power transmitted by Gearing.

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PART IV.—Graphic Statics and Applications to Roofs, Cranes, Beams, Girders, and Bridges.

PART V.—Strength of Materials.—Stress, Strain, Elasticity, Factors of Safety, Resilience, Cylinders, Chains, Shafts, Beams, and Girders.

PART VI.—Hydraulics.—Hydraulic and Refrigerating Machinery.

STEAM AND STEAM ENGINES.

LECTURE I.

CONTENTS.—Early Forms of the Steam Engine: Hero's, Savery's, and Newcomen's.

THE student will find the history of the rise and progress of the Steam Engine both interesting and instructive. Two lectures will therefore be devoted to reviewing, as concisely as possible, the struggles of early inventors to produce mechanical work from steam.

Hero's Engine.—The first application of the elastic force of steam of which there is any record, was by Hero of Alexandria, about 130 B.C.*

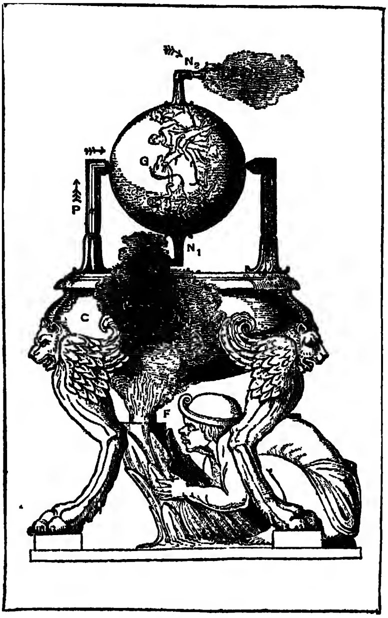
From the following figure and index of parts the construction and action will at once be understood. The fire at, F, heats the water in the caldron, C, generating steam; the steam passes up by the pipe, P, in the direction shown by the arrows into the globe, G, first expelling the air, and then exhausting by the two nozzles, N_1, N_2. Owing to the nozzles being fixed in opposite directions and at right angles to the axis on which the globe is free to rotate, the unbalanced pressures which the steam exerts on that part of each pipe opposite to the opening produce a "couple," and thus turn the globe at a very high speed, but with such a small force that a great expenditure of fuel would be required to develop even a horse-power.

No other notice of the application of steam to produce motion is found until about the year 1563, when Mathesius hints at the possibility of constructing an apparatus similar in its action to that of our modern steam engine.† No device of a

* Glass models, called *Whirling Oelipiles*, are obtainable at any optician's, for illustrating the action of Hero's engine, on the "Barker mill principle."

† For complete descriptions of the attempts made by De Caus, 1624; Giovanni Branca, 1628; Marquis of Worcester, 1663; Sir Samuel Moreland, 1682; Papin, 1685 to 1695, &c., see *Descriptive History of the Steam Engine*, by Robert Stuart, C.E., published in 1825, and dedicated to Dr. Birkbeck, "Patron of the (late) Glasgow Mechanics' Institution, and at one time Professor of Natural Philosophy in the College founded by Professor Anderson in the City of Glasgow." Also, see a treatise by John Farey on *The Steam Engine*, 1827; and Prof. Thurston's *History of the Growth of the Steam Engine*, published by C. Kegan, Paul & Co.

thoroughly practical nature worth drawing the attention of students to occurs, until Captain Thomas Savery brought out his patent steam engine for raising water from mines in 1698.



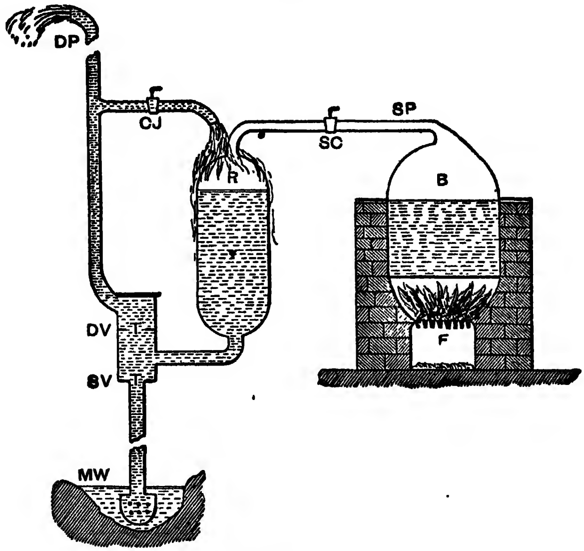
HERO'S ENGINE, 130 B.C.

F for Fire.
C ,, Caldron, containing water.
P ,, Pipe, steam supply.
G for Globe.
N_1, N_2 ,, Nozzles, steam exhaust.

Savery's Engine.—Steam from the boiler, B, is admitted to the receiver, R, by opening the steam cock, S C. When the receiver is filled with steam, the cock, S C, is closed, and C J opened, which allows a douche of cold water to play on the outside of R, thus causing condensation and producing a vacuum. The atmospheric pressure acting on the mine water, at M W, forces water up through the suction valve (or cock), S V, nearly filling the receiver. C J is then closed, and S C opened, thus permitting the steam from the boiler to force the water now in the receiver up through the delivery valve (or cock), D V, and the discharge pipe, D P, to any convenient place clear of the mine.

In Savery's actual engine he adopted a complete duplex set of

boilers, receivers, and cocks, so that the operations of filling one receiver and emptying the other might be conducted simultaneously.* He placed his boilers and receivers about 20 feet above



SAVERY'S ENGINE, 1698.

F	for Furnace.	M W	for Mine water.
B	,, Boiler.	S V	,, Suction valve.
S P	,, Steam pipe.	D V	,, Delivery valve.
S C	,, Steam cock.	C J	,, Condensing jet.
R	,, Receiver.	D P	,, Discharge pipe.

the bottom of the mine water, or well, and the height of the overflow from the discharge pipe about 30 feet above the receiver. The efficiency of a Savery engine, as measured by the weight of coal consumed, was tested by Smeaton in 1774, and found to be about $\frac{1}{10}$ of what can now be realised by a modern pumping-engine. The loss of heat energy, due to the alternate heating and cooling of the receiver, added to the condensation of the steam upon coming into direct contact with the water when forcing the latter out of the receiver, combined with the impossi-

* Desagulier, in 1716, improved upon Savery's engine by introducing a two-way cock between the boiler, the receiver, and the cold water injection, and introduced an inside rose injection for condensing the steam in the receiver. See Stuart on *The Steam Engine*, 1825, Fig. 20.

bility of placing the receiver much more than 20 feet above the bottom of the mine,* and the inability of engineers in those days to construct boilers of sufficient strength to withstand a steam pressure more than 15 lbs. on the square inch,† prevented the adoption of Savery's engine in most mines.

Newcomen's Atmospheric Engine.—In 1705 Thomas Newcomen, a blacksmith, associated with Savery and John Cawley, a glazier, made the experiment of introducing steam under a piston moving in a cylinder. They formed a vacuum by condensing the steam by an affusion of cold water on the *outside* of the steam vessel; and the weight of the atmosphere pressed the piston to the bottom of the cylinder. This was the first form of atmospheric engine—the simplest and most powerful machine that had hitherto been constructed. After a great many laborious attempts at Wolverhampton to make one of their engines work satisfactorily, they were one day (in March, 1712) surprised "to see the engine go several strokes, and very quick together, when, after a search, they found a hole in the piston, which let the cold water in to condense the steam in the inside of the cylinder, whereas before they had always done it on the outside." This fortunate observation gave rise to the improvement of condensing by injection, which thus rendered the cold water jacket of their steam cylinder unnecessary, and they thereafter manufactured their engines in the form shown in the following figure.

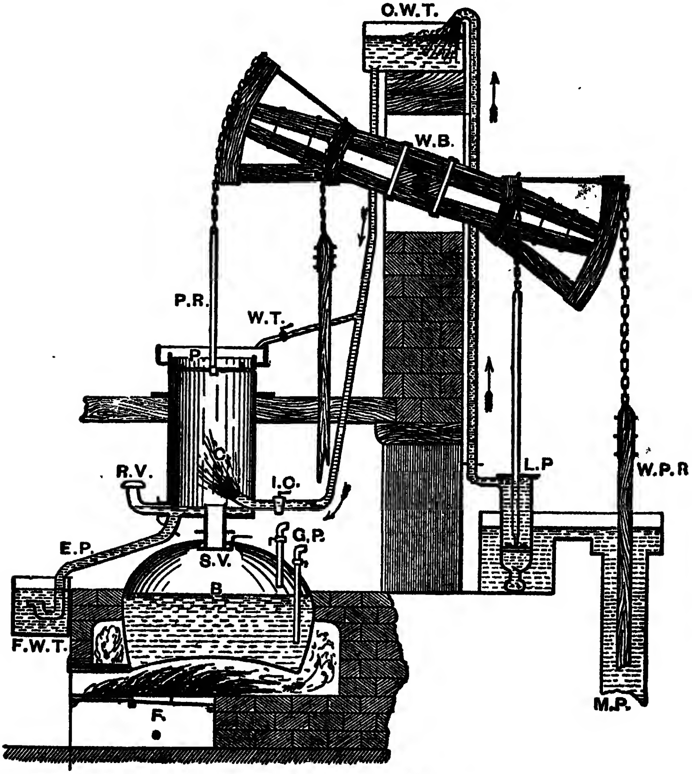
The mine pumps, M P, weighted pump-rod, W P R, and lift pump, L P, on the one side of the wooden beam, W B, being heavier than the piston, P, and piston rod, P R, always brought the piston to the top of the cylinder, C; consequently, to start the engine, the steam valve, S V, was opened, in order to expel the air by the relief or snifting valve, R V,‡ and to fill the whole cylinder with steam. The steam valve was now closed, and the injection cock, I C, opened, which caused a spray of cold water from the cold water tank, C W T, to enter the cylinder and condense the steam. The vacuum produced brought the pressure of the atmosphere into play on the top side of the piston, causing it to descend to the bottom of the cylinder, thus actuating the pumps at the other end of the beam. The condensed steam and injection water got clear away from the bottom of the

* With even a *perfect vacuum* in the receiver, the atmospheric pressure, which is usually about 15 lbs. on the square inch, could only force water up into it from a depth of 34 feet.

† Savery said, "If I could only get boilers and pipes of sufficient strength, I could force water up to a height of 1,000 feet."

‡ This valve was called the *snifting valve* by Newcomen, because the air makes a noise every time it blows through it.

cylinder by the eduction pipe, E P, to the feed water tank, F W T; the water from this tank being used to fill the boiler,



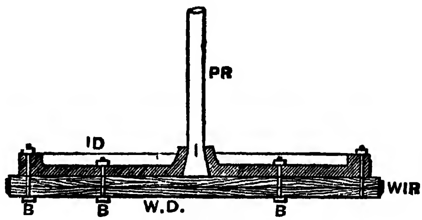
NEWCOMEN'S ENGINE, 1712.

F	for Furnace.	M P	for Mine pump.
B	Boiler.	L P	,, Lift pump.
G P	Gauge pipes.	C W T	,, Cold water tank.
S V	Steam valve.	W T	,, Water tap to top of piston.
C	Cylinder.	I C	,, Injection cock.
P	Piston.	R V	,, Relief or snifting valve.
P R	Piston rod.	E P	,, Eduction pipe.
W B	Wooden beam.	F W T	,, Feed water tank.
W P R	Weighted pump-rod.		

and the height of the water in the boiler was ascertained by the gauge pipes, G P.

At first (in 1712) the valves were opened and shut by hand. To perform these operations at the precise moment, required the most exact and unremitting attention on the part of the attendant, as the least neglect or inadvertence might be ruinous to the engine, by beating out the bottom of the cylinder, or allowing the piston to be drawn out of it. Stops were contrived to prevent both of these accidents; then strings were used to connect the handles of the cocks and valves with the beam, and finally a Mr. Beighton, in 1718, simplified the whole of these movements by causing them to be automatically opened and shut at the proper moment by means of a "tappet rod" connected with the beam. He also introduced the lever safety valve to the boiler.

Another difficulty which at first severely taxed the ingenuity of the inventors was the sudden upheaving of the cylinder, at the moment of creating the vacuum, which caused such a jolt and stress on the pipes connecting the cylinder and the boiler, as to keep them in a chronic state of leakage. It will be observed that at the instant the vacuum is produced, the piston is pressed downwards by the atmospheric pressure, but at the same time the cylinder is equally pressed upwards, so that it required to be very heavy or very securely fastened down, to prevent it rising; since no downward motion of the piston can take place until the inertia of the whole moving mass of beam, pump-rods, &c., has been overcome. This difficulty was in a measure mastered by bolting the cylinder firmly down to strong beams, and keeping it separate from the boiler.



SMEATON'S PISTON.

PR for Piston rod.
ID ,, Iron dish.
WD ,, Wooden dish.
BB for Bolts.
WIR ,, Wrought iron ring shrunk on like a cart-wheel tyre.

Newcomen's piston, which consisted of a flat plate with a broad piece of leather screwed to it and turned up the sides of the cylinder two or three inches, gave considerable trouble,

owing to leakage and the cutting of the leather. An improved form of piston (see preceding figure) was afterwards introduced by Smeaton.*

* For Smeaton's improvements, see Thurston's *History of the Steam Engine;* also, see articles in *The Engineer*, beginning June 6th, 1879, p. 403. For a diagram of a Newcomen engine from an old plate, see *The Engineer*, November 28th, 1879, p. 400, and for a fine diagram of an elaborate Newcomen engine, only taken down in 1880, see the same paper, January 30th, 1880, p. 84. For Mr. Henry Davey's Presidential Address to the Inst. Mech. Engineers on October 16th, 1903, see *Proceedings* of that Institution, and all current Engineering Papers for October 30th and November 6th, 1903.

Lecture I.—Questions.

1. Give a free-hand sketch of Savery's engine, with index of parts. Describe its action in your own words. State clearly how it was so wasteful of fuel, and what limited its application to deep mines.

2. Suppose the water in a mine to be 25 feet below the point to which it rose in Savery's receiver, and the top of the discharge pipe 30 feet above the bottom of the receiver, what vacuum and pressure of steam in pounds per square inch would be necessary to work the engine? *Ans.* 11 lbs., and 13·2 lbs.

3. Give a free-hand sketch with index of Newcomen's engine. Describe in your own words its action, and how you would start it.

4. Suppose the diameter of a Newcomen's engine cylinder to be 30 inches, the stroke 5 feet, the effective pressure per square inch due to the vacuum, 10 lbs., and 15 up and down strokes to be made per minute, how many pounds of water would it lift per minute to a height of 100 feet, neglecting all losses due to friction, &c.? *Ans.* 5303·5 lbs.

LECTURE II.

CONTENTS.—Watt's Model of Newcomen's Engine in Glasgow University—Watt's Single and Double Acting Engines—Hornblower's Engine—List of Steam Engine Patents to 1805.

UP to the period when Smeaton perfected the atmospheric engine, the progress of the "fire engine," as the steam engine was then called, had been merely *empirical;* and in everything that depended on principle, the steam engine of that period was a most rude, wasteful, and inefficient machine. Then came the time when science was to effect more in a few years than mere empirical progress had done in nineteen centuries. In 1759, James Watt had his attention directed by Robison to the subject of the steam engine, and for a few years afterwards made various experiments on the properties of steam. In 1763 and 1764, Watt, while engaged in the repair of a small model of Newcomen's engine (belonging to the University of Glasgow, and since preserved by that University as the most precious of relics), perceived the various defects of that machine, and ascertained by experiment their causes.

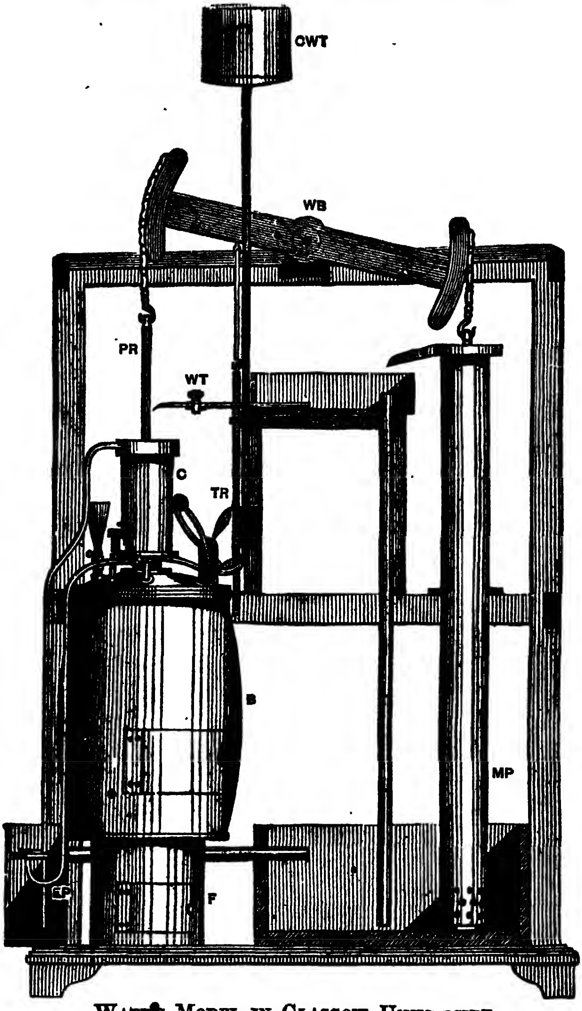
Watt set to work scientifically from the first. He studied the laws of the pressure of elastic fluids, and of the evaporating action of heat, so far as they were known in his time; he ascertained as accurately as he could, with the means of experimenting at his disposal, the expenditure of fuel in evaporating a given quantity of water, and the relations between the temperature, pressure, and volume of the steam. Then, reasoning from the data which he had thus obtained, he framed a body of principles expressing the conditions of the efficient and economic working of the steam engine, which are embodied in an invention described by himself in the following words, in the specification of his patent of 1769:—*

"My method of lessening the consumption of steam, and consequently fuel, in fire engines, consists of the following principles:—

"*First.* That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in

* Extract from *The Steam Engine and other Prime Movers*, by Prof. Rankine.

WATT'S MODEL OF NEWCOMEN'S ENGINE.



WATT'S MODEL IN GLASGOW UNIVERSITY.

F, for Furnace; B, Boiler, C, Cylinder; P R, Piston rod; W B, Wooden beam; M P, Mine pump; T R, Tappet rod; C W T, Cold-water tank; W T, Water tap for keeping piston tight; E P, Exhaust pipe.

NOTE.—In working to repair the model here represented, James Watt, in 1765, made the discovery of a separate condenser, which has identified his name with the Steam Engine.

common fire engines, and which I call the steam vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it; first, by enclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time.

"*Secondly.* In engines that are to be worked wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the steam vessels or cylinders, although occasionally communicating with them; these vessels I call condensers; and, while the engines are working, these condensers ought at least to be kept as cold as the air in the neighbourhood of the engines, by application of water, or other cold bodies.

"*Thirdly.* Whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam vessels or condensers by means of pumps, wrought by the engines themselves, or otherwise.

"*Fourthly.* I intend, in many cases, to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner in which the pressure of the atmosphere is now employed in common fire engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the air after it has done its office.

"*Lastly.* Instead of using water to render the pistons and other parts of the engines air and steam tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver, and other metals in their fluid state."

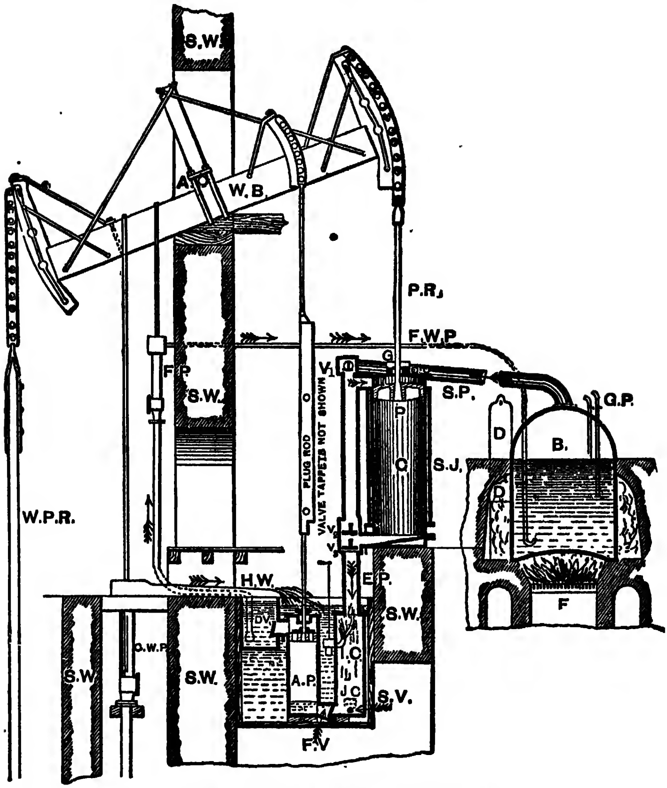
To start Watt's Single-acting Engine:

First. Blow through, by opening all the valves, V_1, V_2, V_3. This allows the steam from the boiler to expel the air from the cylinder, steam passages, and condenser.

Second. Shut valve, V_2, and open injection cock, I C. This creates a vacuum below the piston, and at the same time brings into play the steam pressure above it, causing the piston to descend.

Third. Close valves V_1 and V_3, and open V_2. This allows the steam which forced down the piston to find its way below it, and thus to create an equal pressure on each side of it, when the weight of the pump-rods, acting on the other end of the beam, brings the piston to the top of the cylinder.

These *second* and *third* operations are repeated automatically by the tappet rod when the engine has been fairly started.



• Watt's Single-acting Engine.

F for Furnace.
D ,, Damper.
B ,, Boiler.
F W P ,, Feed water pipe.
G P ,, Gauge pipes.
S P ,, Steam pipe.
V_1 ,, Steam valve.
V_2 ,, Equilibrium valve.
V_3 ,, Exhaust ,,
C ,, Cylinder.
S J ,, Steam jacket.
C C ,, Cylinder cover.
G ,, Gland and stuffing box.
P ,, Piston.
P R ,, Piston rod.

W B for Wooden beam.
A ,, Axis.
W P R ,, Weighted pump-rod down to bottom of mine.
E P ,, Exhaust pipe.
J C ,, Jet condenser.
I C ,, Injection cock.
C W P ,, Cold-water pump.
A P ,, Air pump.
S V ,, Snifting valve.
F V ,, Foot valve.
D V ,, Delivery valve.
H W ,, Hot well.
F P ,, Feed pump.
S W ,, Stone work.

THE FOLLOWING IS AN ABBREVIATED LIST OF IMPROVEMENTS EFFECTED BY WATT ON SINGLE-ACTING ENGINES.

1. Steam jacket to keep cylinder warm.
2. Separate condenser.
3. Air pump to draw off air and condensed steam.
4. Expansive working of steam in the cylinder.
5. Improved piston, cylinder cover, gland, and stuffing box.
6. Cataract or hydraulic governor for regulating the speed.

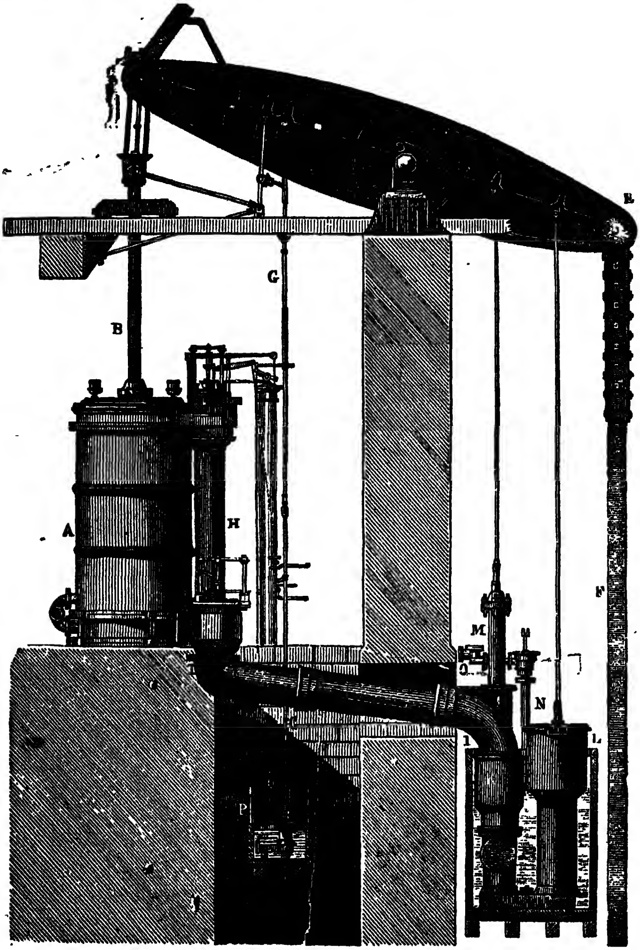
By these several improvements, Watt reduced the amount of fuel required to produce a certain power to about one-third of that required in Newcomen's engine. So fully was this recognised, that Watt, in granting licenses to use his engines, received a *third part of the saving of coals* which was made by his engines, when compared with an atmospheric engine, doing the same work with coals of the same quality.

Watt's Cataract Governor.—The cataract governor was invented by Watt, to regulate the speed of his single-acting pumping engines, and is so simple that the student may readily understand it without the aid of a diagram.* This governor consisted of a pump placed in a tank of water below the level of the cylinder, and the plunger of this pump was attached to a long lever. This lever was loaded with a heavy weight on the same side of the fulcrum as the plunger, and the lever projected out on the other side of the fulcrum. The tappet rod, which was worked off the main beam, engaged with the projecting end of this lever when the piston of the engine was travelling downwards, and therefore raised the plunger of the pump. When the piston of the engine began to rise again (due to the opening of the equilibrium valve by the tappet rod), the heavy weight on the same end of the cataract lever as the plunger caused the latter to descend and to force out the water which it had drawn in during its up-stroke. The water was forced out through a small cock, and the time occupied by the pump plunger in its descent depended upon the amount of opening given to this cock, which could be regulated by the attendant. Since the opening of the steam valve of the engine, which caused the down-stroke of the piston, was effected by a rod from the cataract pump lever, the down-stroke of the engine could not take place until the pump plunger had descended sufficiently to open the steam valve. It will therefore be apparent that by regulating the amount of opening of the discharge cock, the pump plunger could be made to descend with any required

* Large wall diagrams illustrating clearly Watt's cataract governor may be had from the Science and Art Department. It is shown at, P, in the following figure.

speed, and thus the steam valve of the engine opened any required number of times per minute.

The following diagram illustrates an improved form of Watt's single-acting pumping engine, and the cataract governor



IMPROVED FORM OF WATT'S SINGLE-ACTING PUMPING ENGINE.

NOTE.—The student should make a free-hand sketch of the above figure and write out an index of parts, using the first letter of the names of the parts.

Watt's Double-acting Engine.—Hitherto Watt had only introduced steam acting against a piston to press it downwards, thus losing time and the opportunity of taking advantage of the pressure of the steam in the up-stroke to increase the power. In 1782, however, after he had removed from Glasgow to Birmingham, and there joined in partnership with Mr. Boulton, he took out a patent for a "double-acting engine." This engine was freed from the enormous dead weight of counterpoises, which had hung on it from the first attempts of Newcomen, for the purpose of equalising the motion and producing the up-stroke.

Watt says:—"My second improvement upon steam engines, consists in employing the elastic power of the steam to force the piston upwards, and also to press it downwards alternately, by making a vacuum above or below the piston respectively, and, at the same time, employing the steam to act upon the piston in that end, or exerted upon the piston only in one direction, whether upwards or downwards." His 1782 patent engine was considerably improved by his patent of 1784, of which we now give a drawing and description.

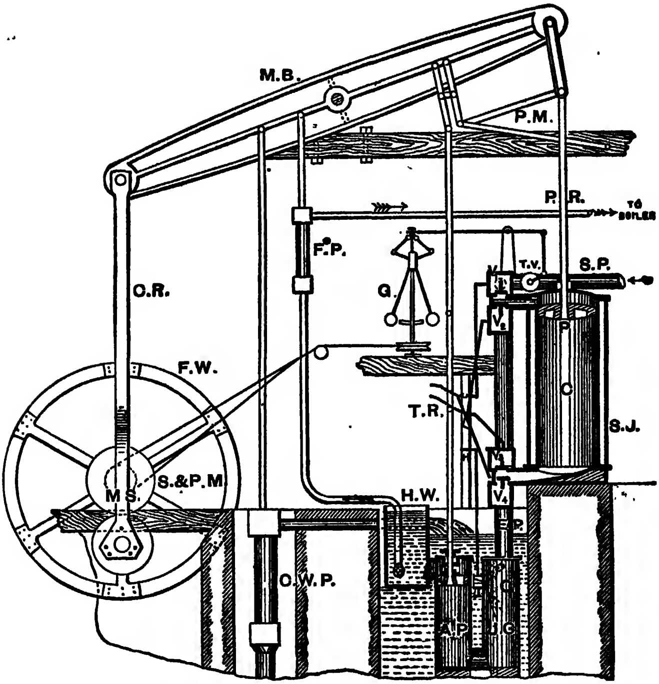
To start Watt's Double-acting Engine:—

First. Blow through, by opening all the valves, V_1, V_2, V_3, V_4.

Second. If the piston is at top of cylinder, shut valves, V_2 and V_3, and open injection cock, I C. This creates a vacuum underneath the piston, and at the same time brings into play the steam pressure above it, causing the piston to descend.

Third. When the piston has reached the end of its stroke, shut valves V_1, V_4, and open V_2, V_3. This permits the steam to exhaust from the top of piston direct to the condenser, and at the same time admits steam from the boiler underneath it, causing the piston to *ascend.*

These second and third operations are repeated automatically by means of the plug rod and tappets.



WATT'S DOUBLE-ACTING ENGINE, 1784.

S P	for	Steam pipe.	H	for	Handle.
T V	,,	Throttle valve.	A P	,,	Air pump.
G	,,	Governor,	H W	,,	Hot well.
V_1, V_3,	,,	Steam valves connected by a pipe.*	F P	,,	Feed pump.
			C W P	,,	Cold-water pump.
V_2, V_4,	,,	Exhaust valves also connected by a pipe.	P	,,	Piston.
			P R	,,	Piston rod.
T R	,,	Tappet (or plug) rod.	P M	,,	Parallel motion.
C	,,	Cylinder.	M B	,,	Metal beam.
S J	,,	Steam jacket.	C R	,,	Connecting rod.
E P	,,	Exhaust pipe.	S & P M	,,	Sun and planet motion.
J C	,,	Jet condenser (separate).	M S	,,	Main shaft.
I C	,,	Injection cock.	F W	,,	Fly-wheel.

* In the drawing the steam pipes connecting valves, V_1 and V_3, and the exhaust valves, V_2 and V_4, cannot be fully shown, but it will form a useful exercise for the student to make a section at right angles to the figure through these valves, including all the necessary pipes.

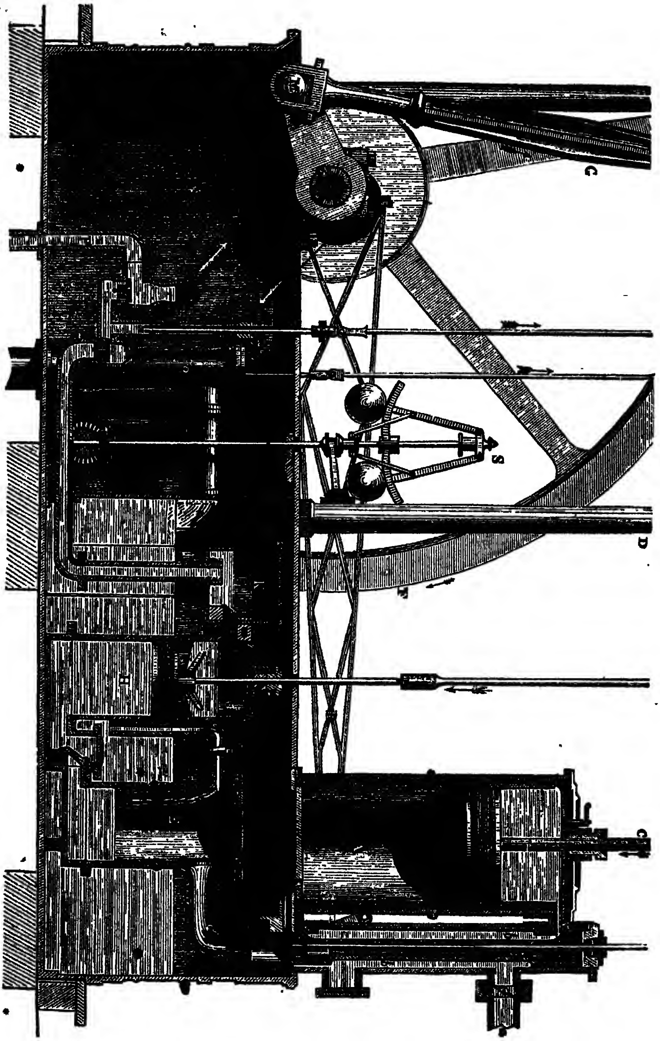
LIST OF IMPROVEMENTS INTRODUCED OR PATENTED BY WATT SINCE THE INVENTION OF HIS "SINGLE-ACTING ENGINE."

1. Applying steam on both sides of the piston.
2. Parallel motion to guide the piston rod in a straight line.
3. Metal beam instead of the large clumsy wooden one.
4. Sun and planet motion to convert reciprocating rectilineal to rotative motion.*
5. Governor to regulate the speed of his rotative engines (see index at end for Governors).
6. Indicator to ascertain the pressure of steam in the cylinder (see index at end for Watt's Indicator).

* There can be no doubt that Watt first thought of applying the crank to convert the reciprocating motion of the piston into a rotative one, but, having neglected to take out a patent, the invention was communicated by a workman to the engineer erecting an engine for a Mr. Matthew Washbrough, of Bristol, who patented the application. The following is Watt's own narrative on this subject:—"Among the many schemes which passed through my mind, none appeared so likely to answer the purpose as the application of a crank in the manner of a common turning-lathe (an invention of great merit, of which the humble inventor and even its era are unknown); but, as the rotative motion is produced in that machine by the impulse given to the crank in the descent of the foot only, and is continued in its ascent by the momentum of the wheel, which here acts as a fly; and, being unwilling to load my engine with a fly heavy enough to continue the motion during the ascent of the piston (and even were a counterweight employed to act, during that ascent, on a fly heavy enough to equalise the motion), I proposed to employ two engines, acting upon two cranks, fixed on the same axis, at an angle of 120° to one another, and a weight placed on the circumference of the fly at the same angle to each of the cranks, by which means a motion might be rendered nearly equal, *and a very light fly* would only be requisite."

It is evident Watt did not then appreciate the advantage of a heavy fly-wheel to equalise motion. The application of a fly-wheel to equalise the motion of the piston was first suggested by Fitzgerald before 1772. Watt, on being informed that his idea of applying the crank to steam engines had been patented by another, said—"In these circumstances I thought it better to accomplish the same end by other means, than to enter into litigation, and, by demolishing the patent, to lay the matter open to everybody."

In order to obtain a rotative motion from a rectilineal one, by some other means than the crank, Watt introduced what is now called the "*sun and planet*" *wheels*, for which he claimed several advantages over the crank, such as more rapid velocity of the fly-wheel, the fly-wheel thus being made to revolve with double the speed that it would in the case of the crank. It is, however, not so simple, while its construction makes it more expensive, and it is easily put out of order; it has now universally given place to the crank. There is a very unique old working model of this "sun and planet" motion, as applied to one of Watt's single-acting engines, and probably made in the end of last century, now in the College of Science and Arts, Glasgow.



IMPROVED FORM OF WATT'S DOUBLE-ACTING ENGINE.

7. Counter for recording the number of strokes of the engine.

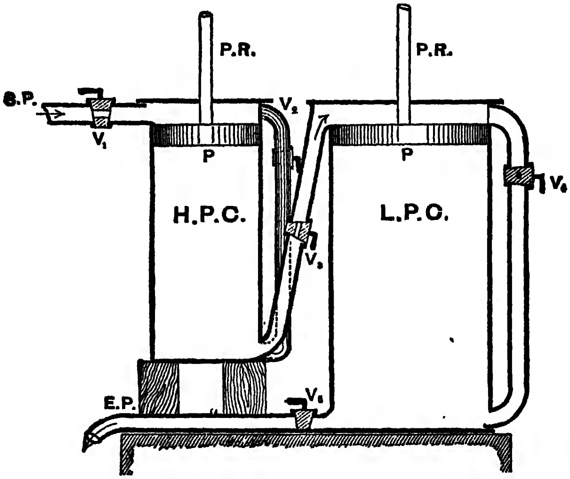
8. Mercury U-tube gauges to ascertain the pressure of steam in the boiler as well as the vacuum in the condenser.

An improved form of Watt's double-acting rotatory engine is shown in the preceding illustration; students should compare this with Watt's engine, shown on page 13.

Hornblower's Engine.—Professor Goodeve says:—"There can be no question as to the fact that Watt invented the expansive working of steam, but, technically, he does not stand first in the records of the Patent Office, for he was anticipated by a patent of Hornblower for a single-acting pumping engine, which dates from the year 1781."

Hornblower, in his specification, says—1. "I use two vessels, in which the steam is to act, and which in other steam engines are generally called cylinders.

2. "I employ the steam after it has acted in the first vessel, to operate a second time in the other, by permitting it to expand



HORNBLOWER'S COMPOUND ENGINE, 1781.*

S P	for Steam pipe from boiler.		V_1, V_3	for Steam cocks or valves.	
H P C	,, High-pressure cylinder.		V_2, V_4	,, Equilibrium	,,
L P C	,, Low- ,, ,,		V_5	,, Exhaust	,,
E P	,, Exhaust pipe to condenser.				

* The drawing is taken from Stuart's *Descriptive History of the Steam Engine*, printed 1825.

itself, which I do by connecting the vessels together, and forming proper channels and apertures, whereby the steam shall occasionally go in and out of the said vessels.

3. "I condense the steam by causing it to pass in contact with metallic surfaces, while the water is applied to the opposite side."

To start Hornblower's Engine:—

First. Blow through (to clear all air and gases out of the cylinders and condenser), by opening all the steam and exhaust valves.

Second. Shut valves V_2 and V_4, and turn on the cold water to surface condenser. This creates a vacuum on the lower side of low-pressure piston, and permits the live steam from boiler to press on high-pressure piston, and at the same time the steam from below that piston to act on the low-pressure piston.

Third. Shut valves V_1, V_3, V_5, and open V_2, V_4. This allows the steam which pressed on the top of each piston to flow underneath them, and thus to create equilibrium when the weighted pump-rods pull them to the top again, ready for another start downwards.

It will be quite apparent to students of the present day, that Hornblower had actually devised not only the compound engine, but also the surface condenser (although his engine was but a single-acting one). He erected several engines on his plan, and, probably, the reason why they did not prove more economical than Watt's single-acting engines, was that the pressure of steam which could be generated in the boilers then constructed was too low. He applied to Parliament, in 1792, for an extension of his patent, but was refused; and it is curious to note the severe criticism of early writers on his invention, the principle of which is nowadays so fully recognised.*

The further improvements on the steam engine by Trevithick, Woolfe, M'Naughton, and others, will be noticed within proper place, in connection with locomotive and marine engines.

In order to complete this Early History of the Rise and Progress of the Steam Engine, we here give a list of a few of the more important English patents up to the beginning of this century.

* Stuart, in 1825, writes—"It must always be a subject of regret, that this ingenious man should have wasted the best part of his life, and ruined his fortune in a series of selfish attempts to copy Mr. Watt's inventions, without coming within the letter of his patent."

* See *The Engineer*, January 28, 1887, p. 70, for a letter discussing the above.

Chronological List of Early Patents.

For Improvements on the Steam Engine, and for Saving Fuel by the Construction of the Fire-Place and Boiler:—

1698.

Thomas Savery, London.

Raising water by the elasticity of steam—Forming vacuum by condensing steam, to raise water by pressure of atmosphere.

1705.

Thomas Newcomen, John Cawley, Dartmouth, and Thomas Savery, London.

Condensing the steam introduced under a piston, and producing a reciprocating motion by attaching it to a lever.

1769.

James Watt, Glasgow.

Invention of the condenser—Use of oil and tallow instead of water—Enclosing cylinder in steam jacket—Moving piston by steam against a vacuum—Steam wheel.

1778.

Matthew Washbrough, Bristol.

Rotative from rectilineal motion.

1781.

John Steed, Lancashire.

Crank movement.

Jonathan Hornblower, Penryn.

Two cylinders.

1782.

James Watt, Birmingham.

Expansive engine—Six contrivances for regulating motion — Double impulse engine—Two cylinders—Toothed rack and sector to piston rod and beam—Semi-rotative engine—Steam wheel.

1784.

James Watt, Birmingham.

Rotative engine—Three parallel motions—Portable steam engine, and machinery for moving wheel carriages—Mode of working hammers and stampers — Improved hand gear—Mode of opening valves.

1785.

James Watt, Birmingham.

Furnace for consuming smoke.

1798.

Jonathan Hornblower, Penryn.

Rotative engine.

1802.

Richard Trevithick and Alexander Vivian, Cornwall.

High-pressure engine.

1804.

Arthur Woolfe, London.

Two cylinders and high-pressure steam boiler.

1805.

James M'Naughton, London.

Saving fuel.

LECTURE II.—QUESTIONS.

All sketches to be done free-hand.

1. Make an outline sketch of the cylinder, piston, and valves connected therewith, in Newcomen's engine; and by the side of it make a second drawing of the cylinder, piston, and valves, as altered by Watt. State briefly the nature of these alterations, and mention the additional parts necessary for the working of Watt's engine, but not shown in your drawing.

2. Explain, with a sketch, Watt's invention of a separate condenser and air pump, as applied to a single-acting steam engine. State the several improvements effected by Watt on Newcomen's engine.

3. What is the principle of the single-acting engine? Draw an outline section through the cylinder and valves, &c. Name the valves and explain their action, also the order of opening and shutting them when starting the engine.

4. In improving the old atmospheric engine, Watt laid down the rule that the cylinder in which the steam did its work should be kept as hot as the steam which entered it. What special provisions did he make for carrying out this rule? Explain your answer by referring to such sketches as may be required.

5. Name the three principal valves connected with the steam cylinder of a single-acting pumping engine. State which are opened and which closed—(1) when the piston is at the top of the cylinder and beginning to descend; (2) when the piston is at the bottom of the cylinder and beginning to ascend.

6. Describe, by a sketch and index of parts, Watt's double-acting engine, and point out the distinction between a single-acting and a double-acting engine. What is the object of the equilibrium valve in a single-acting engine? During what portion of the stroke is this valve open?

7. Enumerate the improvements introduced by Watt into his double-acting steam engine in 1784. Why is this engine so much more economical in steam than the old atmospheric engine?

8. Sketch a section through the cylinders of Hornblower's engine; give index of parts, and state how it is started. Why was the high-pressure cylinder of shorter stroke than the low-pressure one? Wherein is it an improvement on Watt's single-acting engine?

9. Make a vertical transverse section through the nozzles and valves of a Cornish pumping engine, showing the positions of the stop or regulating, steam, equilibrium, and exhaust valves respectively, together with the ports of the cylinder and the passages for the distribution of steam. (*Adv. S. and A. Exam.*, 1889.)

10. Explain, with the necessary sketches, the construction of the cataract of a Cornish pumping engine, and the manner in which it operates upon the valve or valves with which it is connected. (*S. and A. Exam.*, 1890.)

LECTURE III.

CONTENTS.—Temperature—Thermometry—Thermometer Tables—Pyrometry—Pyrometers of Different Kinds, with their Uses, Accuracy, and Ranges—Questions.

It is necessary at the very outset of this section of our subject, to clearly understand what is meant by the different expressions:

1. The *temperature* of a body.
2. The *quantity of heat* in a body, and the *unit of heat*.
3. The *capacity for heat*, and the *specific heat* of a body.

Temperature.—*The temperature of a body is its thermal state considered with reference to its power of communicating heat to other bodies* (MAXWELL).

Two bodies are said to be at the *same* temperature, if, when placed in thermal communication, there is *no* tendency to the transfer of heat between them; if, however, one of them loses heat, and the other gains heat, that body which *gives out* heat, is said to have a *higher* temperature than that which receives heat.

Temperature, therefore, indicates the *quality* or *condition* of the heat in bodies, and is capable of greater or less intensity according to circumstances. It is measured by Thermometers and Pyrometers.

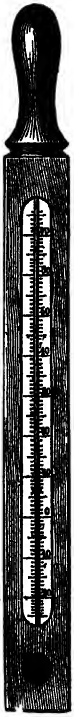
Thermometry.—Thermometry is the method of ascertaining temperatures, or the intensities of heat. The action of thermometers is based on the change of volume, to which bodies are subject with a change of temperature. Air, water, spirit, and mercurial thermometers are severally used under different circumstances, but the mercurial thermometer is the one most commonly employed by engineers. The mercurial thermometer consists of a stem or tube of glass, formed at one end into a bulb, to contain the mercury which expands into the tube. If the stem be of uniform bore, the expansion of the mercury being practically equal for equal increments of temperature, it follows that, if the scale be uniformly graduated, the divisions will indicate equal increments of temperature.*

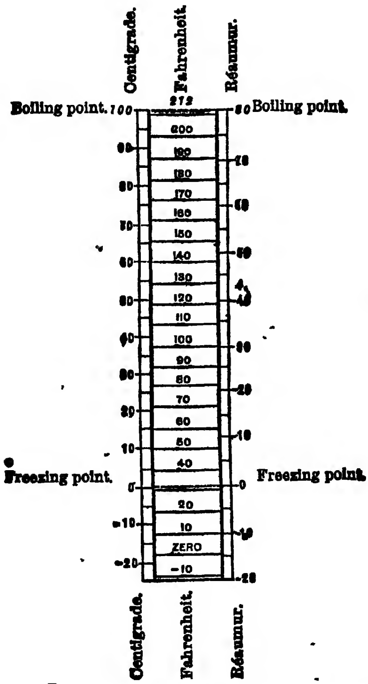
It was early ascertained that the freezing and the boiling points

* For a complete description of the different kinds of thermometers, their construction, calibration, graduation, and use, see Professor Maxwell on "Heat," published by Longman & Co., Text Book of Science Series, Professor Tait on "Heat," published by Macmillan & Co.

of water at the normal pressure of the atmosphere (viz., 14·7 lbs. on the square inch), were constant temperatures, and advantage is taken of this physical property in order to graduate thermometers. The interval between these two fixed temperatures is in the case of the Fahrenheit thermometer (the one commonly used by English engineers) divided into 180 equal parts, termed degrees; in case of the Centigrade,* or standard French thermometer, into 100 equal parts, and in the Réaumur, or the thermometer used in Germany, Russia, &c., into 80 parts. The following comparative scale will render this quite clear:—

Thermometer in Wood Frame.





Thermometer for Screwing into Steam and Water Pipes.

Schaffer and Budenberg's Engineer Thermometers.

* The Centigrade thermometer is now used as the standard thermometer by all the best physicists, and students should familiarise themselves with readings taken by it, as well as with constants and tables to that scale.

Comparison of Different Thermometers.

Fah.	Cent.	Réau.	Fah.	Cent.	Réau.
392	200	160	320	160	128
390·20	199	159·20	318·20	159	127·20
388·40	198	158·40	316·40	158	126·40
386·60	197	157·60	314·60	157	125·60
384·80	196	156·80	312·80	156	124·80
383	195	156	311	155	124
381·20	194	155·20	309·20	154	123·20
379·40	193	154·40	307·40	153	122·40
377·60	192	153·60	305·60	152	121·60
375·80	191	152·80	303·80	151	120·80
374	190	152	302	150	120
372·20	189	151·20	300·20	149	119·20
370·40	188	150·40	298·40	148	118·40
368·60	187	149·60	296·60	147	117·60
366·80	186	148·80	294·80	146	116·80
365	185	148	293	145	116
363·20	184	147·20	291·20	144	115·20
361·40	183	146·40	289·40	143	114·40
359·60	182	145·60	287·60	142	113·60
357·80	181	144·80	285·80	141	112·80
356	180	144	284	140	112
354·20	179	143·20	282·20	139	111·20
352·40	178	142·40	280·40	138	110·40
350·60	177	141·60	278·60	137	109·60
348·80	176	140·80	276·80	136	108·80
347	175	140	275	135	108
345·20	174	139·20	273·20	134	107·20
343·40	173	138·40	271·40	133	106·40
341·60	172	137·60	269·60	132	105·60
339·80	171	136·80	267·80	131	104·80
338	170	136	266	130	104
336·20	169	135·20	264·20	129	103·20
334·40	168	134·40	262·40	128	102·40
332·60	167	133·60	260·60	127	101·60
330·80	166	132·80	258·80	126	100·80
329	165	132	257	125	100
327·20	164	131·20	255·20	124	99·20
325·40	163	130·40	253·40	123	98·40
323·60	162	129·60	251·60	122	97·60
321·80	161	128·80	249·80	121	96·80

Comparison of Different Thermometers—*Continued.*

Fah.	Cent.	Réau.	Fah.	Cent	Réau.
248	120	96	192	88·8	71·1
246·20	119	95·20	191	88·3	70·6
244·40	118	94·40	190	87·7	70·2
242·60	117	93·60	189	87·2	69·7
240·80	116	92·80	188	86·6	69·3
239	115	92	187	86·1	68·8
237·20	114	91·20	186	85·5	68·4
235·40	113	90·40	185	85·0	68·0
233·60	112	89·60	184	84·4	67·5
231·80	111	88·80	183	83·8	67·1
230	110	88	182	83·3	66·6
228·20	109	87·20	181	82·7	66·2
226·40	108	86·40	180	82·2	65·7
224·60	107	85·60	179	81·6	65·3
222·80	106	84·80	178	81·1	64·8
221	105	84	177	80·5	64·4
219·20	104	83·20	176	80·0	64·0
217·40	103	82·40	175	79·4	63·5
215·60	102	81·60	174	78·8	63·1
213·80	101	80·80	173	78·3	62·6
...	...	...	172	77·7	62·2
212	100·0	80·0	171	77·2	61·7
211	99·4	79·6	170	76·6	61·3
210	98·9	79·1	169	76·1	60·8
209	98·3	78·7	168	75·5	60·4
208	97·8	78·2	167	75·0	60·0
207	97·2	77·8	166	74·4	59·5
206	96·7	77·3	165	73·8	59·1
205	96·1	76·9	164	73·3	58·6
204	95·6	76·4	163	72·7	58·2
203	95·0	76·0	162	72 2	57·7
202	94·4	75·6	161	71·6	57·3
201	93·9	75·1	160	71·1	56·8
200	93·3	74·7	159	70·5	56·4
199	92·8	74·2	158	70·0	56·0
198	92·2	73·8	157	69·4	55·5
197	91·7	73·3	156	68·8	55·1
196	91·1	72·9	155	68·3	54·6
195	90·6	72·4	154	67·7	54·2
194	90·0	72·0	153	67·2	53·7
193	89·4	71·5	152	66·6	53·3

COMPARISON OF DIFFERENT THERMOMETERS—*Continued.*

Fah.	Cent.	Réau.	Fah.	Cent.	Réau.
151	66·1	52·8	111	43·8	35·1
150	65·5	52·4	110	43·3	34·6
149	65·0	52·0	109	42·7	34·2
148	64·4	51·5	108	42·2	33·7
147	63·8	51·1	107	41·6	33·3
146	63·3	50·6	106	41·1	32·8
145	62·7	50·2	105	40·5	32·4
144	62·2	49·7	104	40·0	32·0
143	61·6	49·3	103	39·4	31·5
142	61·1	48·8	102	38·8	31·1
141	60·5	48·4	101	38·3	30·6
140	60·0	48·0	100	37·7	30·2
139	59·4	47·5	99	37·2	29·7
138	58·8	47·1	98	36·6	29·3
137	58·3	46·6	97	36·1	28·8
136	57·7	46·2	96	35·5	28·4
135	57·2	45·7	95	35·0	28·0
134	56·6	45·3	94	34·4	27·5
133	56·1	44·8	93	33·8	27·1
132	55·5	44·4	92	33·3	26·6
131	55·0	44·0	91	32·7	26·2
130	54·4	43·5	90	32·2	25·7
129	53·8	43·1	89	31·7	25·3
128	53·3	42·6	88	31·1	24·8
127	52·7	42·2	87	30·5	24·4
126	52·2	41·7	86	30·0	24·0
125	51·6	41·3	85	29·4	23·5
124	51·1	40·8	84	28·8	23·1
123	50·5	40·4	83	28·3	22·6
122	50·0	40·0	82	27·7	22·2
121	49·4	39·5	81	27·2	21·7
120	48·8	39·1	80	26·6	21·3
119	48·3	38·6	79	26·1	20·8
118	47·7	38·2	78	25·5	20·4
117	47·2	37·7	77	25·0	20·0
116	46·6	37·3	76	24·4	19·5
115	46·1	36·8	75	23·8	19·1
114	45·5	36·4	74	23·3	18·6
113	45·0	36·0	73	22·7	18·2
112	44·4	35·6	72	22·2	17·7

COMPARISON OF DIFFERENT THERMOMETERS—*Continued.*

Fah.	Cent.	Réau.	Fah.	Cent.	Réau.
71	21·6	17·3	51	10·5	8·4
70	21·1	16·8	50	10·0	8·0
69	20·5	16·4	49	9·4	7·5
68	20·0	16·0	48	8·8	7·1
67	19·4	15·5	47	8·3	6·6
66	18·8	15·1	46	7·7	6·2
65	18·3	14·6	45	7·2	5·7
64	17·7	14·2	44	6·6	5·3
63	17·2	13·7	43	6·1	4·8
62	16·6	13·3	42	5·5	4·4
61	16·1	12·8	41	5·0	4·0
60	15·5	12·4	40	4·4	3·5
59	15·0	12·0	39	3·8	3·1
58	14·4	11·5	38	3·3	2·6
57	13·8	11·1	37	2·7	2·2
56	13·3	10·6	36	2·2	1·7
55	12·7	10·2	35	1·6	1·3
54	12·2	9·7	34	1·1	0·8
53	11·6	9·3	33	0·5	0·4
52	11·1	8·8	32	0·0	0·0

It is certainly a great inconvenience to have to convert readings taken in one scale to that of another, but students should thoroughly master the simple proportion that exists between the different scales, so as to be independent of conversion tables.*

* Since freezing water, or melting ice, is marked on the different scales as follows:—	32° Fah.	0° Cent.	0° Réau.
and the boiling point of water	212° Fah.	100° Cent.	80° Réau.
we obtain the proportion that exists between the scales by subtracting the freezing from the boiling points, thus	180° Fah.	100° Cent.	80° Réau.

NOTE.—Temperatures as reckoned from the "absolute zero" will be referred to when we come to deal with Pressures and Volumes of Gases.

An easy process in mental arithmetic for converting degrees on the Fahrenheit scale into degrees on the Centigrade scale, and *vice versâ*, is as follows:—

For *Fah. degrees into Cent. degrees.*—Subtract 32 and divide by 2; then add $\frac{1}{10} + \frac{1}{100} + \frac{1}{1000}$ of the result.

Thus, for 60° F. we get $\left(\frac{60-32}{2}\right) = 14$. Then, $(14 + 1{\cdot}4 + {\cdot}14 + {\cdot}014) = 15{\cdot}5°$ C.

Again, for *Cent. degrees to Fah. degrees.*—Multiply by 2 and subtract $\frac{1}{10}$ of the result from the product and then add 32.

Thus, for 15° C. we get $(15 \times 2) - \frac{1}{10}(15 \times 2) + 32$; or, $(30 - 3 + 32) = 59°$ F.

Any desired number of similar examples may be worked out in this way, or by the ordinary proportion and fractional methods as detailed on the following page, and the answers checked by the foregoing tables.

Now, to convert a reading observed on the one scale to its corresponding value on either of the others—

$$\text{Let } F = \text{Temperature Fahrenheit.}$$
$$C = \quad ,, \quad \text{Centigrade.}$$
$$R = \quad ,, \quad \text{Réaumur.}$$

Then we observe that we must *subtract* 32° from the Fah. scale *before* applying the above proportion in converting it to the Cent. or to the Réau., but *add* 32, *after* applying the above proportion, in converting either the Cent. or the Réau. into the Fah. scale, as follows—

$$(F - 32) : C : R :: 180 : 100 : 80$$
$$\text{or as } 9 : 5 : 4$$

$$\therefore \text{ Degrees } C = \frac{(F - 32)\,5}{9}$$

$$,, \quad R = \frac{(F - 32)\,4}{9}$$

$$,, \quad F = \frac{C \times 9}{5} + 32$$

$$,, \quad F = \frac{R \times 9}{4} + 32.$$

Examples.—Suppose the temperature of the feed water for a boiler is 102° Fah., find the corresponding temperature on the Cent. and Réau. scales:

By proportion— $9 : 5 :: (F - 32) : C$

$$\therefore C = \frac{(F - 32)\,5}{9} = \frac{(102 - 32)\,5}{9}$$

$$i.e., \; C = \frac{70 \times 5}{9} = \frac{350}{9} = 38{\cdot}8^\circ \text{ Cent.}$$

Again— $9 : 4 :: (F - 32) : R$

$$\therefore R = \frac{(F - 32)\,4}{9} = \frac{(102 - 32)\,4}{9} = \frac{280}{9} = 31{\cdot}1^\circ \text{ Réau.}$$

Again, by proportion— $5 : 4 :: C : R$

But, $C = 38{\cdot}8^\circ$, $\therefore R = 38{\cdot}8 \times \frac{4}{5} = 31{\cdot}1^\circ$ Réau.

Pyrometry is the method of ascertaining the temperatures of very hot things, as distinguished from *thermometry*, which is really the method of ascertaining the temperatures of warm, hot, or boiling things. Of course, an exact point of demarcation cannot be drawn between the temperatures whereat instruments termed thermometers fail to register heat potential or intensity, and where the other kinds of instruments called pyrometers begin to be applied. In fact, as we shall see later on, the term thermometer is applied to the platinum resistance type of instrument, made for ascertaining temperatures from 14° to 2,500° F. However, the distinction between these two terms is by no means inconvenient or vague, since pyrometers have, generally speaking, been used to measure temperatures beyond the compass of the ordinary mercurial thermometer —*i.e.*, the boiling point of mercury—which is about 650° F. at atmospheric pressure. Further, the word *thermometer* is derived from the two Greek words, θερμός (*thermos*), signifying *warm*, hot, or boiling, and μέτρον (*metron*), *to measure*. In other words, a thermometer is an instrument for indicating the intensity of heat of any *warm* substance. The word pyrometer is also derived from two Greek words, πῦρ (*pyr*), signifying *fire* (or terribly hot), and *μέτρον, to measure*, as before. In other words, a pyrometer is the proper name for an instrument which indicates the intensity of the heat of *very hot substances*. The temperatures of high pressure superheated steam, gas and oil engine exploded mixtures, boiler and superheater furnaces and their flues, dust destructors, kilns, melting and annealing furnaces for different kinds of metals, as well as heating muffles for tempering steel, &c., are all measured by pyrometers. Of late, these instruments have been very much improved, both in regard to their accuracy and their lasting, reliable qualities. So much is this the case, that no truly accurate, scientific investigation into the complete performance of steam, gas or oil engines and electric power plants can be considered correct without their aid. Moreover, the iron, steel, and metal works metallurgist can determine, by their aid, the exact temperatures at which to stop certain operations, in order to obtain the best results; as also, the recently-discovered recalescence stages, or the hiatus positions on the rising and falling scales of temperature at which latent heat is absorbed or given out.

As a practical up-to-date example of the applications of high temperature platinum thermometers, we notice the following quotation from the Blue Book issued by the 1902-03 Naval Boilers Committee of the Admiralty:—"The temperatures of the flue gases were taken by Callendar Electric Thermometers,

and read on a galvanometer made by The Cambridge Scientific Instrument Company, for the special purpose of these trials. The whole temperature-taking apparatus worked satisfactorily throughout. The records were taken regularly from two to four times per hour, as shown in the (Blue Book) tables." It stands to reason, that, if you are able to obtain with comparative ease, and with reliable accuracy, a continuous permanent record of the temperatures of each and every inside and outside part of a set of boilers and engines, during a prolonged trial, you are thereby in a far better position to make a debit and credit balance sheet of the heat generated, usefully applied and wasted, than by the rough and ready methods employed a few years ago. In fact, the pyrometer should now be considered quite as indispensable to the careful, critical, expert engineer as the engine indicator and the dynamo voltmeter.

Pyrometers.—These instruments may be divided into six classes, the first three of which were described in former editions of this book, as well as the fourth and sixth, in the Author's *Manual of Steam and the Steam Engine.* These will therefore be but very briefly noticed here, thus leaving time and space for confining our attention to the two most recent and accurate kinds of electrical resistance and thermo-electric instruments.

First.—Those in which the indications are based upon the change of dimensions of a particular body. For example, Wedgwood's *contracting clay and tapered groove pyrometer*, or Daniell's *expansion metal bar*, enclosed in a black lead case. Neither of these pyrometers are now considered sufficiently accurate.

Second.—Those in which a thin cylinder of platinum, copper, or iron, of known weight and specific heat, are first put into the hot place, whose temperature is to be ascertained, and then immersed into a known weight of water, when the rise in temperature of the latter, as indicated by a mercurial thermometer, enables the temperature of the hot place to be calculated. For example, Wilson's and Siemens' *Water* or *Calorimeter Pyrometers*, which are fairly accurate with care and when new, but they do not admit of more than one temperature being observed at any one time or of the continuous automatic recording of changeable temperatures. The thorough understanding of their construction, action, and manipulation is, however, of considerable educational value to the student.* 6

* Teachers and students may select either copper and iron or any two other convenient, cheap pair of dissimilar metals when demonstrating the principle and action of this pyrometer. They may refer to the Author's *Manual on Steam and the Steam Engine* for a detailed illustrated description of Siemens' Water Pyrometer.

Third.—Those which are based upon the previously-estimated melting or *fusing* points of pure *metals* or metallic alloys. These can only be considered nowadays as rough-and-ready rule-of-thumb aids to workmen, who may also be able from experience to judge, approximately, the temperature of a retort or a furnace by the appearance or colour which it presents to their eyes. For example, dull red was, say, near 1,000° F.; cherry red, 1,450° F.; orange, 2,000° F.; white, 2,350° F.; and dazzling white, 2,700° F.

Fourth.—Those which are based on the fact, that saturated steam or a gas in direct contact with the liquid from which it is generated has the same temperature as the liquid. For example, Schäffer and Budenberg's "Thalpotasimeter or *Pressure Gauge Pyrometer*," as explained in Lecture VII. They are made and graduated to act upon this principle from 100° to 1,400° Fah.

Fifth.—Those which depend on the electrical property of metals, that their resistance increases by a certain amount for a given rise in temperature—for example, Sir William Siemens' Electric Pyrometer and the Callendar-Griffiths *Platinum Resistance* Thermometers. These pyrometers, when connected to a good sensitive galvanometer and battery, are the most accurate and reliable instruments yet devised for indicating very high temperatures up to 2,500° Fah., as will be seen from the following detailed description of their principle, construction and action.

*Sixth.**—Those which depend upon the automatic production of an *electromotive force* or electrical difference of potential or pressure between the two junctions of two different metals connected in series, when these junctions are kept at different temperatures. For example, Le Chatelier's *Thermo-Electric Couple* of platinum to platinum plus 10 per cent. of rhodium, when connected in series with Sir W. C. Roberts-Austen's special device of moving-coil mirror galvanometer combined with a moving sensitised photo-paper, gives good results. Also, Becquerel's platinum to palladium couple, or the platinum to platinum-irridium junction, as used and enclosed in a specially-prepared and refractory porcelain tube by the Cambridge Scien-

* They should also refer to *High-Temperature Measurements*, by Le Chatelier, Boudouard, as translated by Burgess and published in 1901; to the 1893 *Proceedings of the Institution of Mechanical Engineers*, London, for the second report to the Alloys Research Committee, and to their 1902 *Proceedings* for Mr. William Campbell's paper on "Alloys of Copper and Tin," as well as to Munro and Jamieson's *Pocket-Book of Electric Rules and Tables* for the Thermo-Electric Scale, &c., in order to obtain further information upon this interesting and important subject.

tific Instrument Company, in conjunction with a Callendar Recorder, serve to give continuous and permanent records up to 2,000° Fah.

Electrical Pyrometer or Resistance Thermometer.—The most accurate and reliable way of measuring high temperatures is that known as the "platinum resistance method." It is the same in principle as the Siemens Electrical Pyrometer described in the previous editions of this book, but it differs therefrom in several important respects. It was first introduced by Professor Callendar in a paper read before the Royal Society in 1886, and is now supplied in the following form by the Cambridge Scientific Instrument Company:—

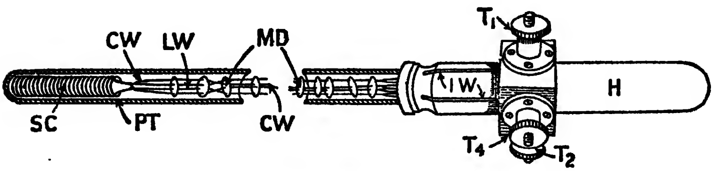


FIG. 1.—PLATINUM RESISTANCE THERMOMETER.

The platinum resistance thermometer consists of a spiral coil of fine platinum wire, the electrical resistance of which varies with the temperature to which it may be subjected. This coil of fine platinum wire, S C, is wound upon a mica frame and is protected from the action of fumes as well as mechanical damage by means of a glass or steel or porcelain tube, P T, according to the temperature to be ascertained. The ends of the coil are fused to two stout platinum or copper leading wires, L W, which are connected at their other ends to two of the four terminals, T_1, T_2, with which the instrument is provided at its cool or handle end, H. Since a change of temperature cannot be confined to the platinum spiral coil, S C, but also affects the leading wires, L W, in the porcelain tube, P T, the latter effect is balanced by constructing the thermometer with a pair of idle or compensating wires, C W. These compensating wires are short-circuited, as shown near the S C coil, and at the other end, H, they are connected respectively to the other two of the four terminals, T_3, T_4. The four leading-in wires are prevented from touching each other in the tube by passing them through holes punched in a set of separated mica discs, M D, which just fit the inside of the tube, P T, and thus, also prevent the passage of convection currents of air along the tube.

Now, referring to figs. 2 and 4, we see, that the terminals, T_3, T_4, of the compensating leading wires, C W, are connected to the opposite arm of the "Wheatstone Bridge" from that arm to which the terminals, T_1, T_2, of the spiral coil, S C, are connected. This arrangement eliminates any error which might be produced by the variation of the temperature of the wires connecting the thermometer with the indicating or recording instrument. The thermometers can thus be placed in positions where it would be impossible to read a mercury thermometer. At the same time, a considerable number of these thermometers may be inserted into different places, flues or furnaces, and distributed over a considerable area. By means of a switchboard, such as that shown in fig. 3, the readings from each of these thermometers may be rapidly and easily ascertained by means of one indicator.

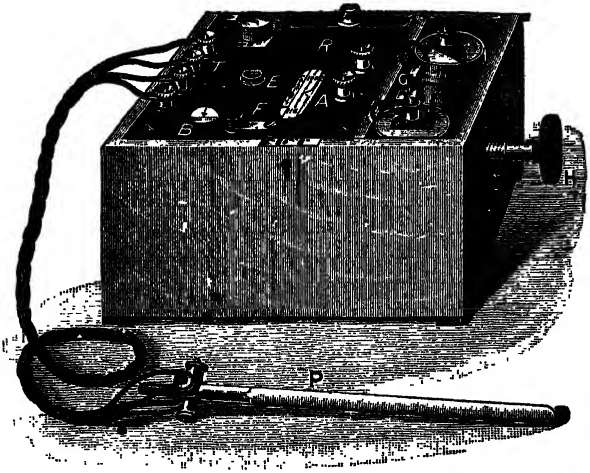


FIG. 2.—CALLENDAR AND GRIFFITH'S RESISTANCE THERMOMETER, P, CONNECTED TO A WHIPPLE TEMPERATURE INDICATOR.

There are two kinds of instruments used with these thermometers for obtaining temperatures, viz. :—Fig. 2, the Whipple Indicator, which reads the temperature directly in degrees Fahrenheit or Centigrade on a galvanometer scale, and Fig. 5, the Callendar Recorder, which not only shows the temperature at any time, but enables a continuous permanent record of the latter to be obtained.

The Whipple Temperature Indicator (fig. 2) consists of a portable moving coil galvanometer, combined with a Wheatstone

Bridge (fig. 4). The resistance of the platinum spiral coil, S C, in the thermometer is balanced by the fixed resistance, R, contained in the instrument, and by the position of the contact, C,

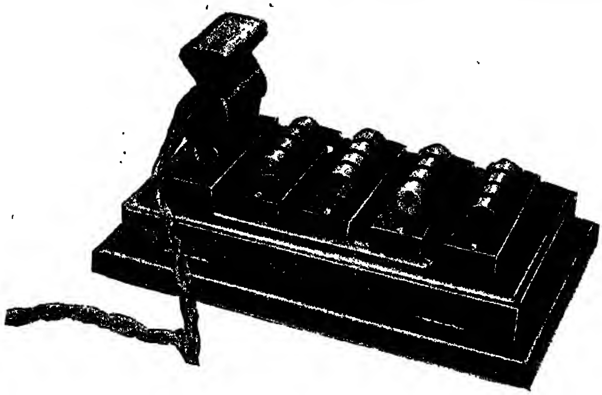


Fig. 3.—Thermometer Switchboard for Enabling Several Different Temperatures to be Taken or Recorded.

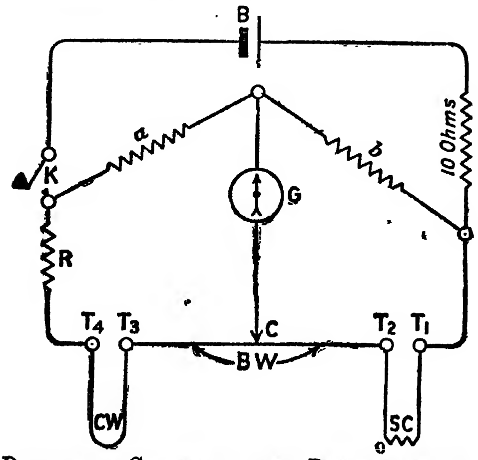


Fig. 4.—Diagram of Connections and Resistances, &c., for the Whipple Temperature Indicator of Fig. 2.

on the balancing wire, B W. The pointer of the galvanometer, G, shows when this balance has been obtained by remaining in the centre opposite to its zero or index mark.

Here, the resistances of the bridge arms, *a* and *b*, are each equal to 10 ohms, or any other convenient equal values. The third arm of the bridge is occupied by a fixed resistance, R, equal to that of the S C coil at 0° Cent. B is a battery of, say, two cells, as shown by fig. 2, with a key, K, for bringing it into action through a resistance of 10 ohms. The balancing wire, B W, is wound in the hollow of a screw thread cut upon an ebonite drum, and the galvanometer contact, C, can be moved round this wire by means of a milled head, H (fig. 2), until a balance has been obtained, as shown by the galvanometer pointer returning to zero. The temperature of S C can then be read off directly from the scale, A (fig. 2), which is connected to the galvanometer contact, C.

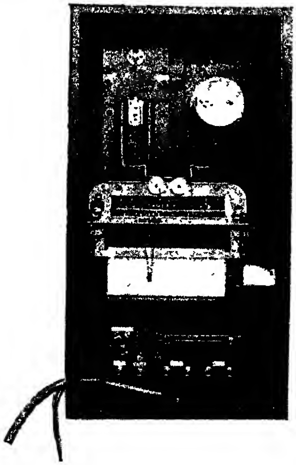


FIG. 5.—CALLENDAR RECORDER, WITH ITS WHEATSTONE BRIDGE, &C.

Callendar Recorder.—In the Callendar Recorder (fig. 5), the method just described is applied, but here the galvanometer contact, C, acts by means of a relay upon an automatically-acting recording pen, which produces a record of temperatures shown by fig. 6. The principle of the recording mechanism is similar to that used in many other instruments, such as the barograph, recording ammeters, and voltmeters. There is the usual cylinder covered with squared paper, divided lengthwise into units of

time and vertically into degrees temperature. It is revolved by clockwork once in two or twenty-four hours. The motion of the pen of the instrument is controlled by electromagnets, whilst the latter are actuated by the differences in the resistances of the bridge balancing wire, B W, on the two sides of the galvanometer contact, C, as shown in fig. 4.

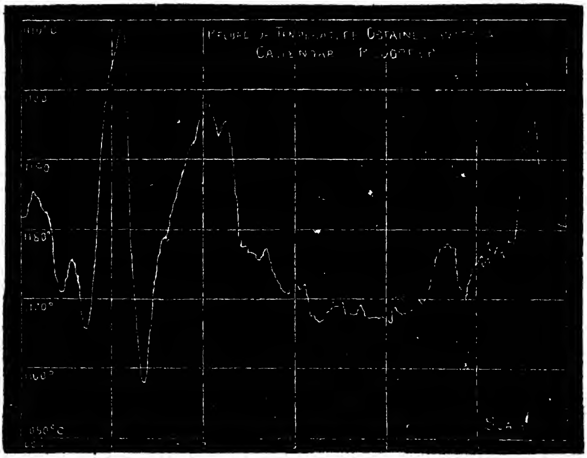


Fig. 6.—Temperature Sheet, as Produced by the Callendar Recorder.

Thermo-electric Thermometers or Pyrometers.—This system of measuring high temperatures was prominently brought before the Institution of Mechanical Engineers and its "Alloys Research Committee," by the late Professor Sir W. C. Roberts-Austen, and is said to be largely used on the Continent for smelting and foundry work.* It is based on the fact, that if two pieces of different kinds of metals are joined together at both ends, and the two junctions are kept at different temperatures, then a difference of electric potential is produced between the two junctions. Consequently, an electric current will pass through the two metals in series. The metals usually employed are pure platinum and some alloy of platinum, such as platinum rhodium or platinum-iridium. These metals are called the "couple." They are fused or twisted together at one end, and

* See the *Proceedings* of this Institution for April, 1893, for description, with illustrations, of his Thermo-electric Recording Pyrometer.

protected by a tube in the same manner as those described for the previous thermometer. The other end of the thermo-couple should be kept as cool as possible and in circuit with some form of galvanometer through which the thermo-electric current passes. This galvanometer may be calibrated to read directly in degrees of temperature. It is also useful to have it calibrated in millivolts, as it can then be standardised at intervals. In the thermo-electric method of measuring temperature, it is advisable to have the galvanometer as close as possible to the couple; for, if long leads are employed, their resistance may introduce an error. The author has found, that one great difficulty often arises with these high-temperature pyrometers, when the heat exceeds 1,800° to 2,000° Fah., from the porcelain tubes breaking, cracking, or bending. In the case of using them for melted brass, the fumes or gases arising therefrom, pass through the cracks or mica lining, and soon destroy the thermo-electric wires and their junctions. He understands, however, that the Cambridge Scientific Instrument Company have recently overcome this difficulty, and are now prepared to submit their instruments to temperatures which were previously considered injurious to accurate pyrometers.

Lecture III.—Questions.

1. Define the temperature of a body. What two natural phenomena have been employed to determine two points of reference in the scale of thermometers? And why?

2. Convert (−461·2° F.), 0° F., 9° F., 32° F., 39·1° F., 60° F., 75° F., 98° F., and 212° F. into degrees on the Cent. scale. Mention what each of these temperatures relate to or are frequently used for.

3. Convert (−274° C.), 0° C., 4° C., 15·5° C., 24° C., 36·6° C., and 100° C. into degrees on the Fah. scale. Mention what each of these temperatures relate to or are frequently used for.

4. Compare the Fah., Cent., and Réau. scales. A Cent. thermometer indicates 15°; show by proportion (in full) how you find what are the corresponding readings in the Fah. and Réau. scales. *Ans.* 59° F.; 12° R.

5. Zinc boils at 1,204° F., mercury at 676° F.; change these readings to Cent. (show your work in full). *Ans.* 651° C. and 358° C.

6. Explain the short methods of converting degrees Fah. into degrees Cent. given in the footnote immediately after the tables in this lecture, and by these convert 200° Fah. into Cent. and 100° Cent. into Fah.

7. Define thermometry and pyrometry. Give their derivations, and explain why the latter term should be employed when referring to temperatures above boiling mercury.

8. Mention the six classes of pyrometers, naming an example of each. State in what cases and why the exact measurement of high temperatures is of value to engineers.

9. Sketch and describe concisely the construction and action of a good platinum resistance pyrometer. How is it used to obtain several different temperatures in different places at the same time?

10. Explain the construction and action of a thermo-electric couple. How is it applied to indicate and to record high temperatures?

11. Sketch and describe the construction and action of an automatic recording apparatus for use with either the platinum resistance or thermo-electric couple pyrometer.

LECTURE IV.

Contents.—Quantity of Heat—British and French Thermal Units—Calorimetry—Bunsen's Calorimeter—Method of Mixture—Definitions of Thermal Capacity and Specific Heat—Examples I. to VI. on Gain and Loss of Heat by Substances, &c.—Specific Heat Table—Thomson's Coal Calorimeter—Rosenhain Form of Thomson Coal Calorimeter—Gas and Oil Calorimeters—Calorific Values of Coal and Gases from Analysis—Specific Heats of Gases and of Steam—Questions.

In the previous lecture the attention of the student was confined to the *first* of the expressions with which it started—viz., the *temperature* of a body and how it is measured. In this lecture the *second* and *third* expressions, *quantity of heat* and *capacity for heat,* will be dealt with.

Quantity of Heat.—The method of measuring the *quantity of heat* in a body is termed Calorimetry,* and the value of that quantity is found in *units of heat.* The expression *quantity of heat* in a body, not only involves a knowledge of the *temperature,* but also of the *capacity for heat* of the body. In fact, the *quantity of heat* in a body is simply the product of its *capacity for heat* and the *temperature* of the body. We have a precise analogy in heat energy, to the way in which two other familiar kinds of energy are estimated—viz., mechanical energy and electrical energy. In the case of mechanical energy, the *quantity of work* put into a body is the product of its *capacity for work* or weight and its *displacement* or distance through which it is moved. In the same way, the *quantity of electricity* put into a body is the product of its *capacity for electricity* and its increase of *potential.* Also, the *quantity of heat* put into a body is its *capacity for heat* into its rise in *temperature.* These three forms of energy are convertible, and may be expressed in ft.-lbs. or units of work.

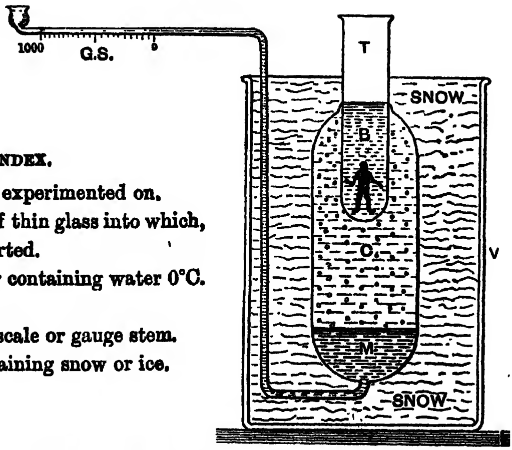
Units of Heat.—To be able to compare different quantities of heat, we must first fix upon a standard substance in a constant condition, and note the effect of raising a unit mass thereof, through our unit of temperature. The most convenient substance is found to be pure water at its maximum density point, and our unit of mass is 1 lb., whilst our unit of temperature is 1° Fah. Since the capacity for heat of water varies so little from 32° to 100° F., any reference to its maximum density point, 39·1° F., or to any other temperature, may be omitted in Engineering questions. Hence, we define the *British Unit of Heat:*—

The British Thermal Unit (symbol, B.T.U.) *is the quantity of heat required to raise* 1 *lb. of water* 1° *Fah.*

* The French Unit of Heat is called the *Calorie,* from the Latin word *Călor,* signifying warmth or heat. It is the quantity of heat required to raise 1 kilogramme of water 1° Cent. It is equal to 3·968 (roughly 4) British units of heat. For small physical and electrical quantities of heat $\frac{1}{1000}$th of this—viz., the small or C.G.S. *Calorie,* or gramme-degree Cent. (gm. 1° C.)—is now universally used as the standard unit of heat.

Calorimetry.—The Ice Calorimeter of Laplace and Lavoisier * consisted of three thin copper vessels of different sizes, so as to permit one being placed inside another. The outer and middle one were packed with broken ice, and were furnished with drain pipes and cocks by which to run off the water from the ice as it became melted. The third or inner vessel held the body to be experimented upon. Although this apparatus furnished good results in the hands of the inventors, it is liable to lead to erroneous determinations, owing to the water produced in the middle vessel adhering to the broken ice, instead of draining completely away.

An improved form of ice calorimeter, designed by Bunsen, is illustrated in the following figure, and is thoroughly reliable in the hands of a good experimeter.



INDEX.

B for Body to be experimented on.
T ,, Test tube of thin glass into which, B, is inserted.
C ,, Calorimeter containing water 0°C.
M ,, Mercury.
G S ,, Graduated scale or gauge stem.
V ,, Vessel containing snow or ice.

BUNSEN'S ICE CALORIMETER.

The body, B, of known weight, which is to give off the quantity of heat to be measured, is first heated in a test tube held in a current of steam of known temperature. It is then dropped quickly into the very thin, dry, clean test tube, T, which is now corked with cotton wool. This test tube is surrounded with solid ice contained in the calorimeter, C. In the bottom of the calorimeter there is a quantity of mercury, M, which extends up through the thin tube to the graduated scale, G S.

* See Maxwell or Tait on "Heat," for a full description of Lavoisier's ice calorimeter and its defects.

The vessel, V, is packed either with newly fallen snow, free from dust particles, or with ice. The ice in the calorimeter is made from distilled water, from which every trace of air has been expelled. If there was air in the water, the process of freezing would expel it, and produce bubbles at the top of the calorimeter, which would vitiate the results; for the accuracy of the test depends upon observing the diminution of the volume of the ice in the calorimeter, C, when a portion of it becomes melted by the heat passing from the body, B. This diminution of volume of a portion of the ice is indicated by the free end of the column of mercury at the graduated scale, G S, moving inwards. The value of these gradations having been previously ascertained, the quantity of ice melted, and consequently the number of units of heat that pass from the body, B, when it has fallen to the temperature of the ice, are easily ascertained.

The value of the gradations on the scale, G S, may be approximately ascertained, by placing a known weight of water at a known temperature in the test tube, T, instead of the body, B, and noting the number of divisions which the free end of the mercury passes inwards, when the water in the test tube has fallen to the temperature of the ice in the calorimeter.

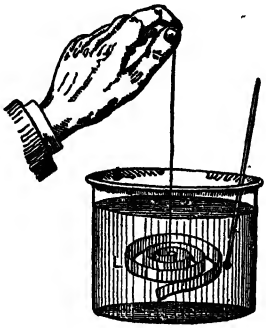
Example I.—Suppose 1 lb. of water at 212° F. to have been placed in the test tube, T, and that, when its temperature had fallen to the temperature of the ice, 32° F., the free end of the mercury at the scale had moved inwards from 0 to 32 divisions on the scale. Now, by placing 1 lb. of lead at 212° F. in the test tube, T, and waiting until its temperature fell to 32° F., if we found that the free end of the mercury only moved inwards by 1 division, we would conclude that the quantity of heat which had passed from the 1 lb. of lead was only the $\frac{1}{32}$ part (·031) of what had previously passed from the water in the test tube to the ice in the calorimeter, under precisely similar circumstances.

The capacity for heat of lead, or its thermal capacity, is therefore $\frac{1}{32}$, or ·031 that of the standard substance—viz., water.

Method of Mixture.—This method depends on the quantity of heat which escapes from one body, increasing the temperature of another body.

To illustrate this method, again take the case of lead. Weigh out 1 lb. of sheet lead, roll it into an open spiral, and attach it to a string. Now, dip the lead into a pot of freely boiling water until it has attained the temperature of the water. While this is going on weigh out a pound of cold water, and ascertain its temperature with a thermometer; say it is 47° F. Then lift the lead from the boiling water, and, while holding it by the string

in the steam rising from the water, allow all water to drop from it, and immerse it quickly in the cold water vessel, keeping it moving by means of the string, so as to bring it intimately into contact with every portion of the water, as shown in the following figure, where, L, is the lead, and, T, the thermometer. Observe the gradual rise in temperature of the water due to the heat passing from the lead, note the point at which it ceases to rise, and suppose that to be 52°F. We have thus ascertained data, from which we may calculate the relative capacities for heat of lead and water, if none of the heat from the lead was given to any other body than to the water.



Thus—The diminution in temperature of the lead from 212° to 52° = 160°; the increase in temperature of the water from 47° to 52° = 5°.

Therefore, since—

The Loss of Heat from the one substance = the Gain of Heat by the other.

Or, the heat from 1 lb. of lead falling 160° = the heat imparted to 1 lb. of water raised 5°.

$$\therefore \frac{\text{The units of heat in 1 lb. of lead}}{\text{The units of heat in 1 lb. of water}} = \frac{5}{160} = \frac{1}{32}.$$

In other words, the capacity for heat of lead is only $\frac{1}{32}$ part that of water, or the same quantity of heat would raise 1 lb. of lead through 32 times as many degrees as it would 1 lb. of water.

Thermal Capacity.—*The capacity for heat, or the thermal capacity of a body, is the quantity of heat required to raise its temperature by one degree.*

The thermal capacity of *unit mass* of a substance is called the *specific heat* of the substance. Hence the definition:—

Specific Heat.—*The specific heat of a substance is the quantity of heat required to raise unit mass of it by one degree in temperature.*

This shows, that the specific heat of water (which is taken as the standard substance) is the same as the "British Unit of Heat" when the temperature of the water is at its maximum

density point. The specific heat of water, however, increases slightly as its temperature is raised, by a mean of $\frac{1}{2}$ of 1 per cent. between the freezing and the boiling points. It is, however, convenient to consider, that a *British unit of heat* and the *unit of specific heat* are identical; because, as we shall see when we come to deal with their corresponding value in units of work, Joule's equivalent of 772 ft.-lbs. of work (or 778 for the latest determination) is the rate of exchange for either or both of these units and *vice versâ*.

The above definition also shows, that the *specific heat* of a substance, is identical with the *ratio* of the *thermal capacity* of any mass of that substance to an equal mass of water. For example, look at the following table of "Specific Heat of Substances," and we see, that the specific heat of ice is (fully) ·5. This means that the capacity for heat of ice at 32° F. is ·5 to 1, or *half* that of an equal mass of water. Again, we see from the same table that the specific heat of lead is ·031. This means (as we have already shown), that the ratio of the thermal capacity of a mass of lead (say 1 lb.) to the same mass of water (viz., 1 lb.) is $\frac{\cdot 031}{1} = \frac{1}{32}$. In other words, a definite weight of water will absorb 32 times the same number of units of heat that the same weight of lead will absorb, in order that the temperature of each may be raised by the same number of degrees.

It is also clear, that if the mass of a body be multiplied by its specific heat and then by the number of degrees of temperature to which the body has been raised or lowered, the combined product must be the total heat units imparted to or withdrawn from the body. Hence:—

Let m = Mass of a substance in lbs.
H_σ = Specific heat (or heat specific) of the substance.
H_T = Total heat units required to raise the temperature of the substance from t_1° to t_2°.

Then, $H_T = m H_\sigma (t_2 - t_1)$ *units of heat*, to raise the temperature of the substance from t_1° to t_2° or to lower it from t_2° to t_1°.

From this statement and formula it is clear, that if one substance receives a certain quantity of heat from another substance, and that this transfer of heat from the one to the other is the only heat which the one gains and the other loses, then:—

$$\left.\begin{array}{c}\textit{The Gain of Heat by the}\\\textit{one Substance}\end{array}\right\} = \left\{\begin{array}{c}\textit{The Loss of Heat by the}\\\textit{other Substance.}\end{array}\right.$$

Hence, Let m_1 and m_2 = Masses of the two substances.
H_{σ_1} and H_{σ_2} = Specific heats of the two substances.
t_1 = Original temperature of m_1.
t_2 = " " m_2.
t_3 = Final temperature of m_1.
t_4 = " " m_2.

Then, supposing that the mass, m_1, receives heat from the mass, m_2, we get :—

$$m_1 H_{\sigma_1} (t_3 - t_1) = m_2 H_{\sigma_2} (t_2 - t_4).$$

If, however, both arrive at the same final temperature, t_3, then,

$$m_1 H_{\sigma_1} (t_3 - t_1) = m_2 H_{\sigma_2} (t_2 - t_3).$$

EXAMPLE II.—We may now apply the knowledge we have gained in this lecture to proving the rule for using Wilson's Pyrometer, as given in our *Manual.* Observe, Wilson plunges a known weight of platinum (for the sake of illustration, assume it to be 1 lb.) at an *unknown* temperature, t_1, into double its weight of water (say 2 lbs.), and notes the rise in temperature, t_2° to t_3°, from which he calculates the original temperature, t_1°, of the platinum, and, therefore, of the furnace from which it had been taken. Referring to the following table, we see that the specific heat of platinum, $H_{\sigma_1} = \cdot0324$, and that of water, the standard substance, $H_{\sigma_2} = 1$, we get the following answer :—

Loss of Heat from Platinum = *Gain of Heat by Water*.

But, *loss* of heat from 1 lb of platinum = $m_1 \times H_{\sigma_1} \times (t_1 - t_3)$.
" " " = $1 \times \cdot0324 \times (t_1^\circ - t_3^\circ)$.
And *gain* of heat by 2 lbs. of water = $m_2 \times H_{\sigma_2} \times (t_3 - t_2)$.
" " " = $2 \times 1 \times (t_3^\circ - t_2^\circ)$.

$$\therefore \quad 1 \times \cdot0324 \times (t_1^\circ - t_3^\circ) = 2 \times 1 \times (t_3^\circ - t_2^\circ).$$

$$\therefore \quad t_1^\circ - t_3^\circ = \frac{2(t_3^\circ - t_2^\circ)}{\cdot0324} = 62\,(t_3^\circ - t_2^\circ).$$

$$\therefore \quad t_1^\circ = 62\,(t_3^\circ - t_2^\circ) + t_3.$$

Or, $\left.\begin{array}{l}\textit{The temperature of}\\ \textit{the platinum}\end{array}\right\} = \left\{\begin{array}{l}\textit{62 times the rise in temp. of the water}\\ \textit{+ the final temp. of the water.}\end{array}\right.$

EXAMPLE III.—2 kilogrammes of mercury at 100° C. are poured into 2·2 lbs. of water at 10° C., what is the temperature of the mixture?

ANSWER—

Let t_1° = Temperature of the mercury.
t_2° = " of water before experiment.
t_3° = " after mixing.

m_1 = Mass of mercury = 2 kilogrammes.
m_2 = ,, water = 2·2 lbs. or 1 kilogramme.
H_{σ_1} = Specific heat of mercury = ·033 (by next table).
H_{σ_2} = ,, water = 1 (,,).

Then, *Loss of Heat from Mercury = Gain of Heat by Water.*

But, *loss* of heat from 2 kilos. of mercury $= m_1 \times H_{\sigma_1} \times (t_1^\circ - t_3^\circ)$.
And *gain* ,, by 1 ,, water $= m_2 \times H_{\sigma_2} \times (t_3^\circ - t_2^\circ)$.

$$\therefore \quad m_1 \times H_{\sigma_1} \times (t_1^\circ - t_3^\circ) = m_2 \times H_{\sigma_2} \times (t_3^\circ - t_2^\circ).$$
$$2 \times \cdot 033 \times (100 - t_3^\circ) = 1 \times 1 \times (t_3^\circ - 10^\circ).$$
$$6 \cdot 6 - \cdot 066\, t_3^\circ = t_3^\circ - 10^\circ.$$
$$6 \cdot 6 + 10 = t_3^\circ + \cdot 066\, t_3^\circ.$$
$$16 \cdot 6 = 1 \cdot 066\, t_3^\circ.$$
$$t_3^\circ = 16 \cdot 6 \div 1 \cdot 066 = 15 \cdot 5^\circ \text{ C.}$$

SPECIFIC HEAT OF SUBSTANCES.

BY REGNAULT AND OTHERS.

From D. K. Clark's "Rules, Tables, and Data," at between 32° and 212° Fah., unless stated.

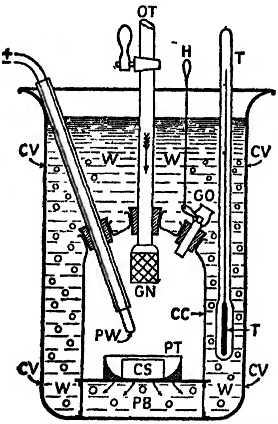
Water at 39·1° F.,	1·000	Silver,	·057
,, 212° F.,	1·013	Platinum, sheet,	·0324
Ice at 32°,	·504	,, spongy at 952° F.,	·035
Steam at 212°,	·480	Coal,	·240
Mercury,	·033	Coke,	·200
Iron, cast,	·130	Olive oil,	·310
,, wrought,	·113	Air,	·238
Steel, soft,	·116	Carbonic oxide,	·248
Copper,	·095	,, acid,	·217
Lead,	·031	Hydrogen,	3·404
Zinc,	·093	Oxygen,	·218
Tin,	·057	Nitrogen,	·244

Note 1.—Students will find it to be both interesting and instructive to try and verify (even roughly), by means of a home-made Bunsen calorimeter, any of the specific heats in the above table. Before doing so, however, it will be advisable to study the detailed description of how to ascertain with great accuracy the complete constant for the gauge tube graduated scale, G S, and then for the correction due to the simultaneous melting and forming of ice in the calorimeter, C, owing to its being surrounded by a freezing mixture (see previous figure and former editions of this book).

Note 2.—The specific heat of an elementary solid (pure body) is inversely as its atomic weight, or the specific heat multiplied by the atomic weight is a constant quantity.

Coal Calorimeters.—It is very important that engineers should have a simple, ready and accurate instrument, whereby they may test the correct values of the heat-producing qualities of different kinds of coal. In all complete and exact trials of steam plant, it is of the utmost importance to ascertain the pounds of water converted into steam per lb. of coal burned in the furnace. This proportion does not, however, give a definite idea of the actual heat units per lb. of coal consumed, since an indefinite amount of the total heat evolved may have been lost through radiation and conduction to surrounding bodies, as well as through the flues and chimney. In order to find out whether the boiler is doing its duty, it is necessary to test samples of the coal independently of the boiler. This can only be done by the aid of a good calorimeter.

William Thomson's Coal Calorimeter.—In previous editions, Mr. Lewis Thompson's Calorimeter or Fuel Tester was illustrated and described; but owing to the fact, that 10 per cent. had to be added to its indicated heat units, in order that a more or less accurate approximation



INDEX TO PARTS.

C V for Calorimeter vessel.
W ,, Water.
T ,, Thermometer.
C C ,, Combustion chamber.
P B ,, Perforated base.
P T ,, Porcelain tray.
C S ,, Coal sample.
P W ,, Platinum igniting wire.
+, — ,, Leads to battery.
O T ,, Oxygen tube.
G N ,, Gauze nozzle.
G O ,, Gas outlet.

DIAGRAM OF THE WILLIAM THOMSON COAL CALORIMETER OR FUEL TESTER.

might be arrived at, this method of igniting and burning the coal sample had to be improved, as well as the exact determination of the specific heat of each element of the apparatus. Mr. William Thomson, Analytical Chemist, the Royal Institution Laboratory, Manchester, has given great attention to this subject and brought out an improved calorimeter by keeping these special purposes in view. A diagrammatic or educational view of his apparatus, with an index to parts, is shown by the figure:—It consists of a glass vessel, C V, into which is placed a Woulff's bottomless three-necked bottle, resting upon a perforated metal or porcelain base. On this base is placed a porcelain or platinum tray, P T, containing the coal sample, C S, to be burned. This sample is ignited by pushing down the tube containing the + and — leading wires from an ordinary battery. The lower ends of these wires are connected to a platinum igniting wire, P W. Whenever this wire comes into contact with C S, the battery circuit is closed and the platinum wire becomes white hot, thus setting fire to the sample whose calorific value has to be ascertained. In order to

keep up rapid combustion and to ensure that every particle of the sample is thoroughly consumed, a flow of oxygen under pressure is turned on to C S from the oxygen tube, O T. The lower end of this tube terminates in a cylindrical wire gauze nozzle, G N, to spread the oxygen and thus prevent the too-rapid breaking-up or splitting of the sample. When the sample has been apparently consumed, as seen through the surrounding glass vessel, C V, containing the water, W, and the combustion chamber, C C, it is finally stirred up by the platinum wire, P W, in order to burn the very last iota. Then, the supply of oxygen is turned off and the gas outlet tube, G O, is opened by pulling up the handle, H, connected by a metal wire to this outlet cock, as shown by the accompanying figure. This permits the head of water, W, in C V to press through the perforated base, P B, and to fill the combustion chamber, C C, so as to bring the water into intimate contact with the gaseous products of combustion (which are shown rising as bubbles through the water), as well as with everything that has been heated by the combustion of the small coal sample.

The gradual rise in temperature of the whole of the water, W, is now noted by taking readings on the sensitive thermometer, T, which is graduated to about half-an-inch per degree Fah. into tenths and one-hundredths of a degree. The times in seconds are also noted by a stop watch, so that the *highest mean temperature* reached by the water may be very exactly determined. This temperature is taken as the value, with which to make the following simple calculation of the *calorific value*, C_V, of the coal sample in British thermal units (B.T.U.). If all the heat which is generated by the burning of the sample be communicated equally throughout to the water and to the several things contained in it, and, if the highest mean temperature to which these attain be exactly noted, as well as the equivalent value in "grammes of water." Then :—

The Heat Units given out by Sample = The Heat Units absorbed by Water, &c.

Or, $$C_V \times w = W \times t,$$

i.e., The calorific value, $C_V = \frac{W t}{w}$ gm.-deg.-Fah.

Where, w = Weight of coal sample in grammes.

W = Weight of water + equivalent weight of water of the other things in it in grammes.

t = Maximum rise of mean temperature of W in, say, degrees *Fah.*

Note.—The following table gives the figures obtained for one of the William Thomson Coal Calorimeters, as used for the following calculations :—

Material Used.	Weight in Grammes.	Specific Heat of Material.	Equivalent to Grammes of Water.
Glass of beaker = 7·812 ozs.,	221·472	·1977	43·784
Glass bell,	48·015	·1977	9·492
Brass,	106 017	·09391	9·956
Iron,	12·993	·11379	1·478
Platinum,	7·3496	·03244	·238
Clay support,	16·875	·1977	3·336
India-rubber,	1·184	·2	·237
Mercury,	27·192	·0333	·905
Thermometer glass,	4·161	·1977	·822
Copper gauze,	27·122	·09515	2·581
Water employed,	..	..	2,000·000
Total material heated in the calorimeter equivalent to water,			2,072·829

EXAMPLE IV.—Suppose that the coal sample, w, weighed 2 grammes, that W, the weight of water, and the other things in it had the equivalent shown by the previous footnote—viz., 2,072·8 grammes—and that the mean maximum rise of temperature of W was 12·7° Fah.; find the calorific value, Cv, of the coal sample.

From the previous equation and formula, and substituting the given values, we get—

$$C_V = \frac{W t}{w}.$$

Or,
$$C_V = \frac{2072 \cdot 8 \times 12 \cdot 7}{2}.$$

$$\therefore \quad C_V = 13{,}162 \textit{ gramme degrees Fah.}$$

Hence, it follows that each gramme of such coal could give out 13,162 gm.-Fah.° of heat; or, that *each lb.* of such coal if perfectly burned in a boiler furnace, and if the *whole of its heat of combustion* were transmitted to the water in the boiler, the water would receive 13,162 lb.-Fah.°, or B.T.U. of heat.

Now, if it be desired to know what weight of boiler water this quantity of heat would evaporate, or convert into steam at atmospheric pressure, we have only to know that water boils at 212° Fah. under these circumstances, and that 966 B.T.U. are absorbed in converting every lb. of the water into steam; or, that the latent heat of steam (as will be seen later on) is 966 B.T.U. Hence:—

$$\text{Weight of water evaporated} = \frac{13{,}162}{966} = 13 \cdot 6 \textit{ lbs.}$$

Of course, we do not get this splendid result in actual daily practice, even from the very best Welsh coal and with the most perfect boiler ever made, but it is the aim and object of every good engineer to get as near to it as he can. In most cases, as we shall see later on, 10 to 12 lbs. of water evaporated from and at 212° Fah. per lb. of the good coal, having a calorific value of about 14,000 B.T.U., is considered good work.

The Rosenhain Form of Thomson Coal Calorimeter.—The student should now have no difficulty in understanding the construction and action of the latest form of this instrument, as made by the Cambridge Scientific Instrument-Making Company, by aid of the following figures, index to parts, and concise description, as it involves no further principles than those just enunciated

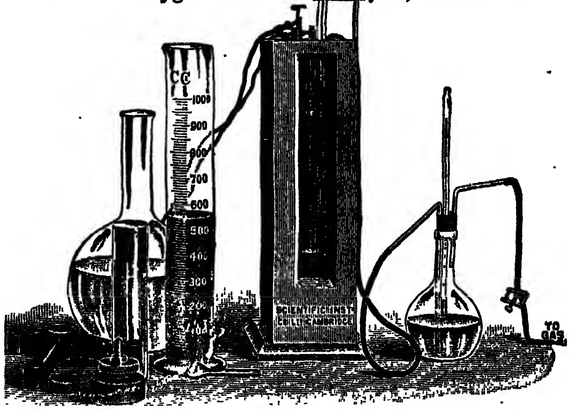
Construction and Manipulation of the Instrument.—As in the former apparatus, this instrument consists essentially of two main parts, viz.:—A polished brass box (instead of a deep glass jar), with a bottom and two diametrically opposite glass windows. This forms the containing vessel, C V, holding the water, W, and the combustion chamber, C C, in which the coal sample, C S, is burned. The combustion chamber, C C, consists of an ordinary glass lamp-chimney, closed at the top and the bottom by brass clamping plates, C P_1 and C P_2, with rubber washers, R W.

When the platinum or porcelain tray, P T, containing the coal sample, C S, has been placed on the plate, C P_2, and the three upright rods, U R, have been inserted into their lugholes in C P_2, then C C is put upon the lower R W, and the upper plate, C P_1, with its attachments, is laid on the upper end of the combustion chamber. Three brass nuts, N, are then screwed upon the three upright rods, thus drawing the two end clamping plates firmly into contact with the top and bottom rubber washers, R W.

A ball joint, B J, containing a stuffing-box, S B, is mounted on the upper plate, CP_1. Through this stuffing-box there passes a leading wire tube, L T, containing two wires connected at the upper end of W_1, W_2 (as shown by the first outside view), with the + and — terminals of a suitable electric battery, giving about 6 volts, which is used to render incandescent the platinum wire, P W, connected to the lower clips of W_1, W_2. This platinum wire is for igniting the coal sample, C S; and, as will be readily understood, it can be pushed down or pulled up through S B, so as to bring it into contact with C S, or to remove it therefrom.

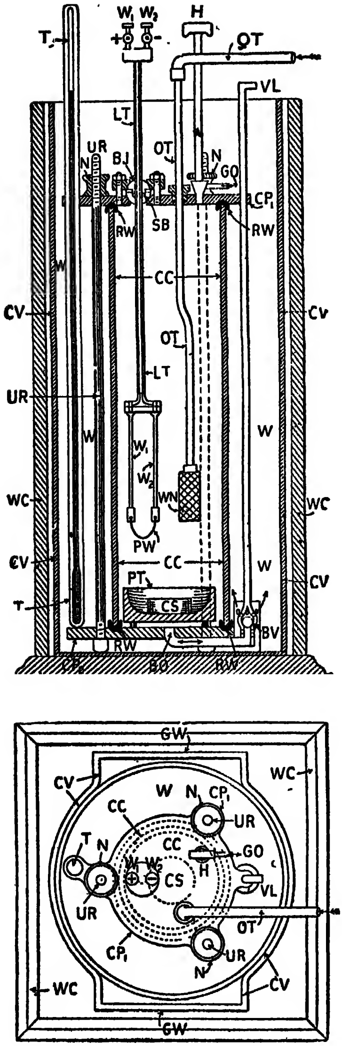
The upper plate, CP_1, also carries an oxygen tube, O T, connected by its outer end to an oxygen supply under pressure. At its lower end it is covered by a wire gauze nozzle, W N, for the purpose of supplying and gently distributing the necessary oxygen to the sample (when the latter has been ignited by the heated platinum wire), in order to thoroughly burn up the coal or other fuel under test, without fracture, as previously explained.

The upper plate, CP_1, also carries a gas outlet, G O, which is worked by a cock and handle, H, so that, if it be desired to let the water, W, into the combustion chamber, C C, through the bottom inlet or outlet, B O, after the sample has been consumed and after the oxygen is cut off, a little of the gas



ROSENHAIN FORM OF THOMSON'S COAL CALORIMETER.
Made by the Cambridge Scientific Instrument Company.
Connected up to Battery and Oxygen Supply, and Ready for Testing.

products of combustion may readily escape by G O. Then the natural head of water, W, and the pulling up of the bottom valve, B V, by the valve-lifter handle, V L, permits the surrounding water to enter the combustion chamber, C C, through B O and flood the whole interior of C C. At the same time, the products of combustion in C C partly escape by B O when G O is closed, and the whole of the water which has entered C C may be forced out into C V by the application of oxygen from O T, so that the water in C V can be brought into intimate contact with everything that has been heated by the burning of C S.

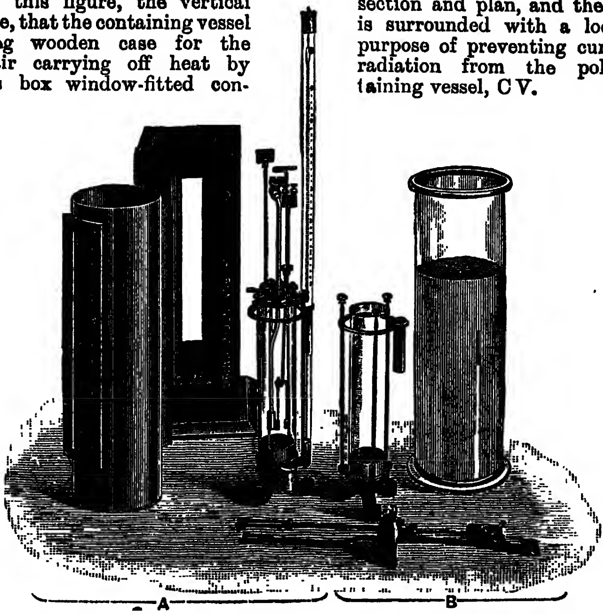


Index to Parts.

W C for Wooden casing or outside cover.
C V ,, Calorimeter vessel
G W ,, Glass windows for C V.
C S ,, Coal sample.
P T ,, Porcelain tray.
C C ,, Combustion chamber.
$CP_{1,2}$,, Cover plates for top and bottom of C C.
R W ,, Rubber washers for C C.
U R ,, Upright rods.
N ,, Nuts for U R.
B J ,, Ball joint.
S B ,, Stuffing-box.
L T ,, Leading-in wire tube.
W_1, W_2 ,, Wires to P W.
+, — ,, Leads to battery.
P W ,, Platinum wire igniter.
O T ,, Oxygen tube.
W N ,, Wire gauze nozzle.
B O ,, Bottom outlet or inlet to chamber, C C.
B V ,, Ball valve for bottom outlet and inlet.
V L ,, Valve lifter for B V.
G O ,, Gas outlet.
H ,, Handle for G O.
T ,, Thermometer.
W ,, Water in C V covering the combustion chamber, C C.

Vertical Section and Plan, with Index to Parts, of the previously Illustrated Rosenhain-Thomson Coal Calorimeter.

The thermometer, T, is now read as previously described in explaining the Thomson instrument. This thermometer is graduated into degrees Centigrade and subdivisions, so that 0·01°. C. may be correctly read. Another thermometer, graduated to 0·01° C., is also used for the purpose of taking the temperature of the atmosphere and of the oxygen supply in the oxygen wash-bottle, as seen to the extreme right of the first figure; whilst the ordinary water supply bottle and a 1,000 c.c. measure, for holding the necessary water is seen on the left of that figure. It will be observed from this figure, the vertical section and plan, and the next figure, that the containing vessel is surrounded with a loosely-fitting wooden case for the purpose of preventing currents of air carrying off heat by radiation from the polished brass box window-fitted containing vessel, C V.



ROSENHAIN-THOMSON COAL CALORIMETER, showing Forms A and B Made by the Cambridge Scientific Instrument Company.

This casing is dispensed with in the cheaper and simpler form, B, shown on the right hand of the above figure. Also, in this form, the products of combustion, the aperture or bottom outlet, B O, communicates directly with the water, W, without the intervention of a ball valve, B V, for the gas pressure in C C, can be made sufficient to keep out or let in the water, W, as required.

Accuracy of the Instrument.—Full instructions how to use these calorimeters are supplied by the makers with each instrument. They state, that the complete combustion of a small compressed cylinder of coal takes from 7 to 15 minutes, according to its weight, by aid of this instrument, whilst less than ½ per cent. of the sample escapes being thoroughly burned, and that no carbon monoxide need be formed when the supply of oxygen, &c., is properly regulated. It is worth noting here, however, that after

the maximum reading of the thermometer, T, has been observed (by taking the values at stated short intervals after the water has been finally expelled from the combustion chamber), the entire instrument is allowed to cool, with a slight current of oxygen passing through it for a period of time equal to half of that which has elapsed between the commencement of the combustion and the maximum reading of the thermometer. Then, the fall of temperature during this time is added, *as a radiation correction*, to the apparent rise of temperature observed between the initial and maximum readings of the thermometer.

Here, in this instrument, we see, that the thermometers used are graduated to the Cent. scale, and that the weight of the coal specimen is taken in grammes, whilst the times are observed in seconds. Hence, everything is noted in accordance with the truly scientific and now universal centimetre-gramme-second system of carrying out accurate physical or electrical experiments.

EXAMPLE V.—

Let $W = 3{,}270$ gms. of water (for the whole instrument, as before).

$w =$ 1·425 gms. for weight of coal specimen.

$t =$ 3·34° C. for apparent rise of W + ·08° C. for radiation correction.

Then, since the calorific value, Cv, must be in gramme degrees Cent., or French calories, we get, by the same reasoning and formula as before—

$$C_V = \frac{W t}{w} = \frac{3{,}270 \times 3{\cdot}42}{1{\cdot}425} = 7{,}850 \text{ C.G.S. calories.}$$

But, since a degree Cent. is equal to $\frac{9}{5}$ of a degree Fah., we have only to multiply 7,850 by 9 and divide by 5 in order to get the result, 14,130, in B.T.U.

Gas and Oil Calorimeters.—In view of the fact, that gas and crude mineral oils are now frequently burned instead of coal for generating steam in boilers, as well as for producing power in gas engines, it is important that engineers should be able to measure accurately their calorific values. The principle and the action of calorimeters adapted for this purpose will be readily understood from what has been stated in this lecture. The heat generated by the flame of the burning gas or oil is transmitted to a current of water flowing at a constant rate in a somewhat similar way to that of a surface steam condenser. Then, measurements are simultaneously taken of—

(1) The quantity of gas or oil burned in a certain time;
(2) The quantity of water passed through the calorimeter in the same time;
(3) The constant or mean difference of the temperature in degrees of the water on entering and leaving the apparatus during the experiment.

Junkers' Gas and Oil Calorimeter.—The general arrangement of the whole apparatus, as set up and ready for a gas test is shown by Fig. 1 with its index to parts. A vertical section and sectional plan through the calorimeter vessel is explained by Fig. 2 and its index to parts. Finally, Fig. 3 shows a special burner for testing the calorific value of oils, spirits and other liquids.

Testing Gases.—From the general view in Fig. 1 and sectional views in Fig. 2, it will be seen, that the gas to be tested is measured for quantity by a gas meter, G M, for temperature by a thermometer, T_1, and for pressure by a meter, P M, and gauge, P G, before it enters the burner, B, in the calorimeter vessel, C V. The heat produced from the flame, F, which rises

from the burner, strikes the inside of the combustion chamber, C C, and the heated gases, H G, turn round near the top of this chamber and enter the upper ends of a series of vertical tubes surrounded by water. These gases flow down the tubes and give up their heat to the water before issuing by the spent gas outlet, G O.

The cold water is led by the water-supply pipe, W S, to an elevated cistern, from which it gravitates through the C W pipe to an adjustable tap, A T, where its temperature is taken by the thermometer, T_2. It then flows upwards and around the numerous heated gas tubes, H G, inside the calorimeter vessel, C V. The heated water, H W, is thus forced up through a series of divisional or disc plates, D P, in order to thoroughly mix it and enable the thermometer, T_3, to register the temperature which it has attained from the heated gases. The hot water overflow, H W O, then passes into a soil-pipe funnel until the difference of temperature, as found by T_2 at the inlet and at the outlet by T_3, becomes constant, when it is turned into a water measure, W M. The other and smaller water measurer, W M, is for the purpose of collecting any condensed vapour from the inside of the containing vessel, C V. For every cubic centimetre of water collected in this smaller vessel an allowance of 0·6 calorie must be made and deducted from the gross value, as shown by the example.

Testing Oils.—The only difference between the arrangements for testing the calorific values of gases and oils, or other liquids, lies in the burner. For this purpose a special arrangement has been provided, as shown by Fig. 3. The liquid is contained in an oil reservoir, O R, which has a screwed top or nipple, *n*, connected to a force air-pump for the purpose of driving the liquid up to the burner, B. The whole of this special burner and its fittings can be suspended from one arm of a balance whilst the flame is playing up inside the combustion chamber, C C, of the calorimeter previously described. By taking off a weight from the scale pan side equal to the desired amount of oil to be burned, the balance will show when this quantity has been consumed by its pointer arriving at zero of its scale, then the experiment can be stopped and the calorific value of the consumed oil ascertained with the same accuracy, and in the same way as now to be described for gases.

EXAMPLE VI.—

Let C_V = Calorific value obtained from the burned gas or liquid.

„ W = Weight of water passed through the apparatus and heated.

„ t = Temperature difference of inflow and outflow water.

„ G = Gas or oil burned during the test.

Then, $C_V = \dfrac{W\,t}{G}$ calories or heat units per unit of gas or oil burned.

Suppose that the following results were obtained :—

Gas Meter.	Cold Water by T_2.	Hot Water by T_3.	Water Passed.
0·344 [illegible]	8·77° C.	26·77° C.	2 litres.

Then, $G = 0·344$ cubic foot ; $t = (T_3 - T_2) = (26·77 - 8·77) = 18°$ C. ;
$W = 2$ kilogrammes.

Hence, $C_V = \frac{Wt}{G} = \frac{2 \times 18}{\cdot 344} = 104\cdot 65$ (large) calories per cb. ft. of gas burned.

Now, if it has been found, that 2 cubic feet of gas, when burned, caused 53 c.c. of water to become condensed and drained into the small water

INDEX TO PARTS.

G I for Gas inlet.
$T_{1, 2, 3}$,, Thermometers.
G M ,, Gas meter.
G P ,, Gas pipes.
P M ,, Pressure meter.
P G ,, Pressure gauge.
B ,, Burner.
C V ,, Calorimeter vessel.
G O ,, Burnt gas outlet.
W S ,, Water supply.
W O ,, Water overflow.
C W ,, Cold water supply.
A T ,, Adjusting tap.
D C ,, Drain cock.
H W O ,, Hot water outlet.
W M ,, Water measures.

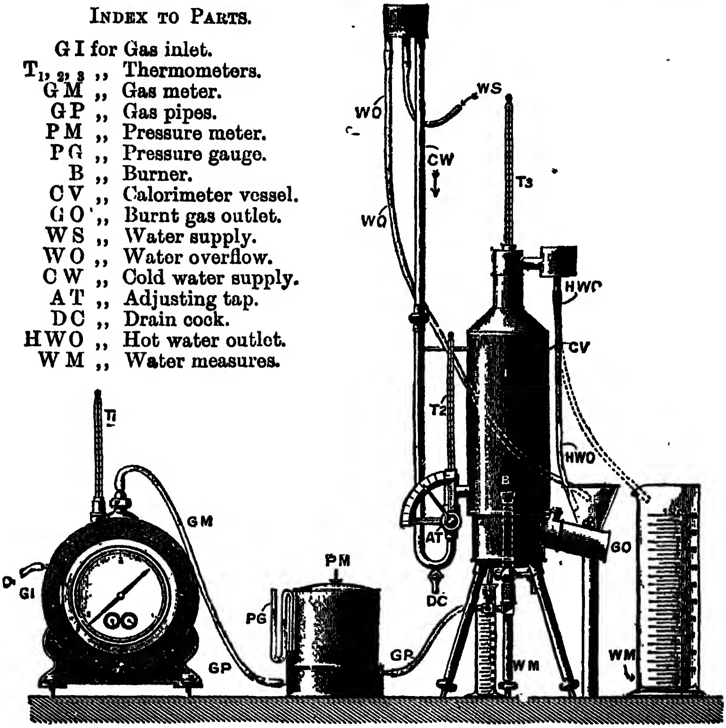


FIG. 1.—JUNKERS' GAS AND OIL CALORIMETER.
In Complete Working Order.
(By Hermann Kühne, Limited, London.)

measure, W M, its calorific value per cubic foot of gas consumed will be, as previously explained :—

$$\frac{0\cdot 6 \times 53}{2} = 15\cdot 9 \text{ calories.}$$

Hence, the net calorific value per cubic foot of the gas in the present instance will be—

$$(104\cdot 65 - 15\cdot 9) = 88\cdot 75 \text{ (large) calories.}$$

And, $\frac{88\cdot 75 \times 9}{5} = 160$ B.T.U. (approximately).

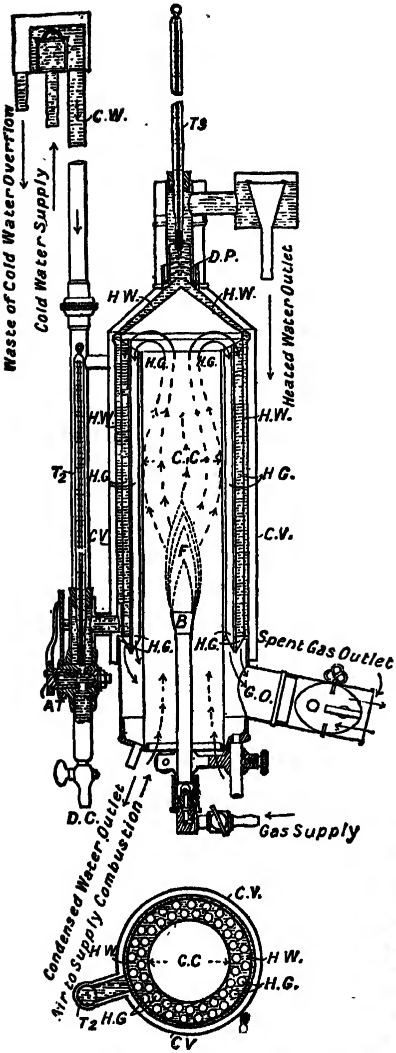


FIG. 2.—VERTICAL SECTION AND PLAN OF JUNKERS' CALORIMETER.

INDEX TO PARTS.

For Fig. 2.

C W for Cold water.
A T ,, Adjustable tap.
D C ,, Discharge cock.
T_2 ,, Inlet thermometer.
C V ,, Calorimeter vessel.
H W ,, Heated water.
D P ,, Disc plates.
T_3 ,, Outlet thermometer.
B ,, Burner.
F ,, Flame.
C C ,, Combustion chamber.
H G ,, Hot gases passing down through the vertical tubes.
G O ,, Gas outlet for spent gases.

For Fig. 3.

B for Burner.
O R ,, Oil reservoir.
n ,, Nipple.

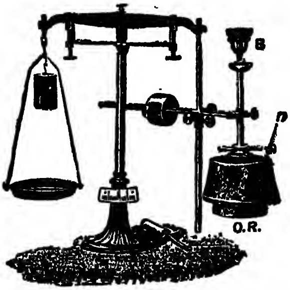


FIG. 3.—SPECIAL BURNER FOR TESTING OIL FUELS WITH JUNKERS' CALORIMETER.

Calorific Values of Coals and Gases from their Chemical Analysis.—If we obtain a correct analysis of any coal, gas, or oil and refer to a table of the heat units per lb. or per gm. for each element contained therein, we can calculate the approximate calorific value of the coal, gas, or oil, but this method of arriving at the result is now giving way to the more practical and direct calorimeter measurement just described.*

EXAMPLE VII.—Taking the following formula, as used by Messrs. Brame and Cowan in their "Comparison of Different Types of Calorimeter," and applying the same to their analysis of a sample of coal, where—

Carbon (C)	= 90·09 %.	Hydrogen (H)	= 3·85 %.
Sulphur (S)	= ·77 ,,	Ash . .	= 1·68 ,,
Oxygen and nitrogen (O+N)	= 3·61 ,,		

We get the calculated calorific value, Cv, in large calories by the formula:—

$$C_V = \frac{1}{100}\left[8{,}140\,C + 34{,}500\left\{H - \frac{(O+N)-1}{8}\right\} + 2{,}220\,S\right].$$

$$C_V = 8{,}567 \text{ calories} = 15{,}421 \text{ B.T.U.}$$

This high value was only 0·7 per cent. lower than their best result by experiment with the Mahler Calorimetric Bomb, but it was 0·9 per cent. higher than their greatest with the William Thomson Calorimeter.

Specific Heats of Gases.—It is important, and in fact necessary to distinguish between the specific heat of a gas at constant pressure and its specific heat at constant volume.

The specific heat of a gas when kept at constant pressure is the quantity of heat required to raise unit mass thereof one degree in temperature. In this case, the gas is considered in the same way as that in which we defined the specific heats of liquids and solids. For example, the specific heat of perfectly dry pure air under this condition is 0·2377, or approximately ·238, at all temperatures and pressures.

The specific heat of a gas when kept at constant volume is the quantity of heat required to raise unit mass thereof one degree in temperature. Under this condition the specific heat of perfectly dry pure air is represented by the number 0·1688, or approximately 0·17.

The ratio of these two specific heats for dry air is $\frac{\cdot 238}{\cdot 17} = 1{\cdot}4$,

and this ratio is practically the same for the other gases which cannot be readily condensed into liquids. It may also be considered as approximately true, in regard to both these ways of

* See different books on Gas and Oil Engines, such as Bryan Donkin's, published by Charles Griffin & Co., and Professor Perry's *Steam, Gas, and Oil Engines*, under "Combustion and Fuel." Also see "Comparison of Different Types of Calorimeter," by J. S. S. Brame and Wallace A. Cowan, in No. 22, vol. xxii., of the *Journal of the Society of Chemical Industry*, November 30, 1903.

estimating the specific heats of gases *which cannot be readily liquefied*, that :—

(1) The specific heat of a definite gas is the same at all temperatures and pressures.

(2) The specific heats of different gases are inversely as their densities, when the latter are compared at the same temperature and pressure. Or, the thermal capacities of equal volumes of different gases are equal at the same temperature and pressure.

Specific Heats of Steam.*—As stated in the footnote to Table II., Lecture VII., on the "Properties of Dry Saturated Steam," the specific heat of superheated steam is usually taken at Regnault's estimate of 0·48. His experiments consisted in determining the total heat necessary to raise water from 32° F. or 0° C. to temperatures of about 120° C., and to 220° C. *under the constant pressure of the atmosphere*, then taking the differences of these two experiments as being the heat necessary to raise water from 120° C. to 220° C. This involves the assumption that steam of 20° C. (or 36° F.) above the boiling point is in the condition of steam gas. Later researches indicate, that the specific heat of steam for a small amount of superheat is greater than for the higher superheats now adopted with steam engines.

Experiments are being conducted at present by the Reichs Anstalt, Charlottenburg, Berlin, and by the British National Physical Laboratory, but their results were not ready for publication when this 14th Edition was issued.

* See *Mém. Acad. Sci.*, vol. xxvi., pp. 170, 909. Also, *Proc. Lit. and Phil. Soc. of Manchester*, 1897; and *Scientific Papers*, by Prof. Osborne Reynolds, F.R.S., vol. ii., p. 65, on "Methods of Determining the Dryness of Saturated Steam and the Condition of Steam Gas." See also *Report of the British Association for* 1897, p. 554, on "The Specific Heat of Saturated Steam," by Prof. J. A. Ewing, F.R.S., and Prof. S. Dunkerley.

LECTURE IV.—QUESTIONS.

1. What do you mean by the quantity of heat in a body, and how is it measured?

2. What is the unit of heat adopted in Great Britain? How many units of heat are imparted to a cubic foot of water (62·5 lbs.), on raising it from 60° to 212° F., also to 1 lb. of copper? *Ans.* 9,500, and 14·44.

3. Define and show the difference between the terms "capacity for heat" and "specific heat" of a substance. Suppose a substance was given to you to find its specific heat, how would you conduct the experiment? Give an arithmetical example.

4. If 1 lb. of platinum is plunged into 1 lb. of water at 50° F., and the resultant temperature of the water is 112° F., what was the original temperature of the platinum? *Ans.* 2,025·5° F.

5. If 2 lbs. of copper at 500° F. are plunged into 4 lbs. of water at 60° F., what will be the resulting temperature? *Ans.* 80° F.

6. Define "specific heat." Deduce a formula for determining the relation between the masses, specific heats, &c., when two substances are mixed together. A piece of platinum, weighing 1 lb., is suspended in the hot gases of a furnace whose temperature has to be ascertained. After being heated to the temperature of the furnace it is taken out and plunged into 2 lbs. of water at 49° F. The resulting temperature of the mixture is found to be 100° F. Determine the temperature of the furnace, having given specific heat of platinum = 0 034. *Ans.* 3,100° F.

7. Sketch and describe the principle and action of Thomson's coal calorimeter. Explain clearly, why the "water equivalent" of each item therein which is subjected to heat from the burnt specimen of coal must be accounted for if accurate results are to be obtained by this instrument. Show how these are arrived at and how the total "water equivalent" is computed.

8. The total "water equivalent" of a Thomson's coal calorimeter is 10 lbs., the weight of the coal specimen is 0 01 lb., and the maximum rise in temperature of the water, &c., is 10° F. What is the heat value of the specimen in B.T.U. per lb. of coal and in calories per kilogramme? Calculate how many lbs. of water every lb. of this coal would convert into steam at and from 212° F., if the combustion was perfect and if all the heat therefrom entered the water.

9. Sketch and describe concisely the construction and action of the Rosenhain-Thomson coal calorimeter. If the "water equivalent" in this case be 4 kilogrammes, weight of specimen 2 grammes, apparent rise in temperature of water 3·9° C., and the radiation correction 0·1° C.; what is the calorific value of the coal in C.G.S. calories and in B.T.U.? What weight of steam would this coal raise at and from 100° C.?

10. Define the two ways of reckoning the specific heat of gases. Is the specific heat of a gas supposed to be the same at all temperatures and pressures? How does the specific heats of different gases vary with their densities? What do you know about the specific heats of wet, saturated, and superheated steam?

11. Sketch and describe Junkers' calorimeter, and explain how it is used for ascertaining the calorific values of gases, oils, or other combustibles.

12. Explain and give an example of how the calorific values of combustibles may be obtained from their chemical analysis.

13. How do we determine approximately the calorific value and the quantity of air required for the complete combustion of any combustible gas of which we know the chemical composition? What is your notion of the construction of a contrivance which would enable us to measure the calorific value? A coal contains 84 per cent. of carbon, 6 per cent. of hydrogen, 1 per cent. of oxygen. What is its calorific value? Take the calorific value of carbon as 14,500 and of hydrogen 4·28 times that of carbon. How much water at 60° F. will 1 lb. of this fuel convert into steam at 212° F.? (S. & A., 1897, Adv.)

14. Given the following analyses of different samples of coal. Calculate, by aid of the formulæ in this lecture, their respective calorific values in calories and B.T.U.:—

Samples.	C.	H.	S.	Ash.	O + N.	Answers in Calories.
B, . in °/o,	81·02	3·23	0·64	9·50	5·61	7,527
C, . in °/o,	87·79	4·09	0·59	3·14	4·39	8,425
D, . in °/o,	84·07	4·51	0·685	5·69	5·045	8,241
E, . in °/o,	78·29	4·76	1·48	4·90	10·57	7,638

15. What are the ultimate constituents of a steam coal upon which the value of the fuel as a heat-producer depends? Describe the important chemical actions which take place during the combustion of coal, and obtain an expression for the *calorific value* of the fuel, and for the amount of air required theoretically and in practice for the complete combustion of a coal containing given proportions of carbon, hydrogen, and oxygen; hence determine how many lbs. of water at 62° F. could be theoretically evaporated into steam at 212° F. by the complete combustion of 1 lb. of a coal containing 84 per cent. of carbon, 5 per cent. of hydrogen, and 1 per cent. of oxygen, and what would be the minimum weight of atmospheric air that would be necessary to completely burn each lb. of such a coal? (S. & A., 1897, Hons.)

16. A pound of fuel contains:—Carbon, 0·886 lb.; hydrogen, 0·041 lb.; and oxygen, 0·028 lb. What is its calorific value without deducting for the latent heat of the steam produced? If, in a *perfect* boiler the gaseous products weigh 12 lbs., their average specific heat being 0·238, and the boiler steam is at 341° F. while the boiler-room is at 60° F., what percentage of the whole heat is necessarily carried away? If the feed is at 60° F., how many lbs. of steam would be produced by a perfect boiler? If a common boiler at the same pressure produces 9 lbs. of steam per lb. of fuel, what is its efficiency? (B. of E., 1898, H., Part i.)

17. In a boiler trial, a continuous collection is made of samples of the furnace gases as they leave the boiler, and a volumetric analysis of the samples collected gives the following figures:—CO_2 = 10·35 per cent.; O = 8·10 per cent.; N = 81·55 per cent. The coal used during the trial has 87·3 per cent. of carbon, 3·7 per cent. of hydrogen, 1·4 per cent. of oxygen, 2·3 per cent. of nitrogen, and the rest is ash. Find how many pounds of air have been admitted to the furnace per pound of coal burnt. Also find, given that the air temperature during the tr[illegible] was 61° F., and the temperature of the escaping furnace gases 753° F. [illegible]e loss in thermal units in the waste gases per pound of coal burnt. The specific heat of CO_2 = 0·217, of O = 0·218, and of N = 0·244. (C. & G., 1901, H., Sect. B.)

LECTURE V.

CONTENTS.—Transfer or Diffusion of Heat—Radiation—Conduction—Convection—The Ebullition and Circulation of Water in Steam Boilers—Questions.

Transfer or Diffusion of Heat.—It was explained in the last lecture, that equality of temperature between two bodies exists, when there is no tendency to a transfer of heat from either to the other. We saw also that, when their temperatures differed in the slightest degree, there is a tendency to an equality of temperature, by a transfer of heat from the hotter to the colder, and that this tendency is greater, the greater the difference of temperature between the bodies.

Rankine states that the rate at which the transfer of heat takes place between two bodies, at unequal temperatures, depends—

"*First.* On the tendency to transfer heat, increasing as some function of the two temperatures and their difference.

"*Secondly.* On the areas of those parts of the surfaces of the bodies through which the transfer of heat takes place. In most of the cases which occur in practice, those areas are equal, and then the rate of transfer of heat is directly proportional to their common extent.

"*Thirdly.* On the nature of the material of each of the bodies, and the condition of their surfaces.

"*Fourthly.* On the nature and thickness of the intervening substances, if any. Increase of that thickness diminishes the rate of transfer of heat.

"The transfer of heat takes place by three processes, called respectively, *radiation*, *conduction*, and *convection*.

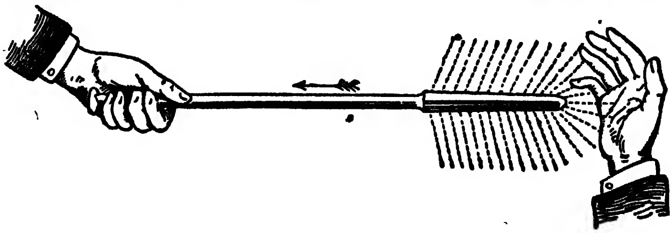
"Radiation of heat takes place between bodies at all distances apart, in the same manner and according to the same laws with the radiation of light." *

Radiation.—To illustrate the radiation of heat from one body to another, take a common poker, heat it to redness in the fire, and hold one hand a few inches from the heated end, as shown in the figure.

The hand experiences the sensation called heat, owing to the transfer of the same in straight lines from the hot poker, as it were by radial vibrating rays of heat energy.

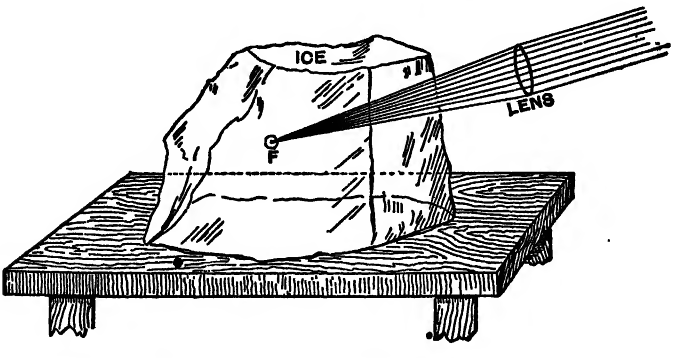
* From Rankine on *The Steam Engine*, p. 257.

Another common, but interesting illustration, is that of making a convex lens of ice, by pressing a heated concave scale-pan of a balance on a block of ice, and holding this lens between the sun and your coat at the proper distance, so as to



focus the heat rays on the same. The lens of ice as well as the air will be scarcely affected by the heat rays passing through them, while the coat will soon be burned.

An even still more interesting and striking experiment, due to Professor Tyndall, is that of focussing the heat rays from the sun or a strong electric arc light on the interior of a block of ice.



The heat rays pass through the mass of ice without apparently affecting it, except at the point where they meet; here the ice very soon becomes melted.

The phenomenon of radiation consists, therefore, in the transmission of energy from one body to another by propagation through the intervening medium, in such a way that the progress of the radiation may be traced, after it has left the first body and before it reaches the second, travelling with a certain velocity and leaving the medium behind it in the condition in which it

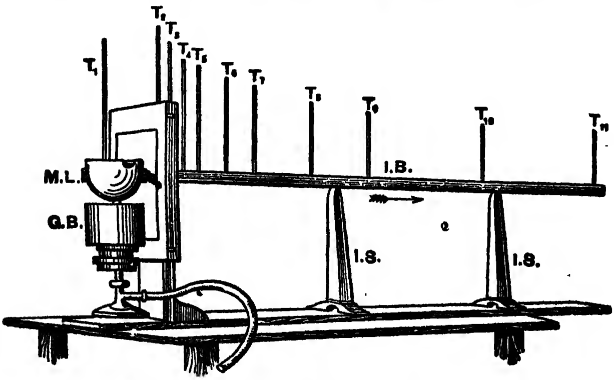
all of the same thickness and superficial area, and subject them all on the one side to a certain temperature, and on the other side to the same number of degrees more or less.

DEFINITION.—*The thermal conductivity of a body at any temperature is the number of units of heat which pass, per unit of time, per unit of surface, through an infinite plate (or layer) of the substance, of unit thickness, when its sides are kept at temperatures respectively half a degree above and half a degree below that temperature* (TAIT).

Although the above definition is perfect, in as far as it lays down theoretically a thoroughly systematic way in which the relative conducting powers of different substances may be compared, it is found practically impossible to realise experimentally such simple conditions.

The methods chiefly employed for measuring thermal conductivity depend ultimately upon observations of the temperature of the body at different parts of its mass.

The temperature effects of a given quantity of heat are inversely as the capacity for heat of the body; hence, what is directly deduced from such experiments is not the thermal conductivity as just defined, but *its ratio* to the capacity for heat of the body.* Thus, these experiments require in addition, the



FORBES' EXPERIMENT ON CONDUCTIVITY.

I B for Iron bar.
I S ,, Insulating supports.
$T_1, T_2 \ldots T_{11}$,, Thermometers.
M L for Melted lead.
G B ,, Gas burner (Argand).

* This is termed *Thermometric Conductivity* by Maxwell and DIFFUSIVITY by Sir William Thomson (see tables at end of Conduction).

determination of the specific gravity and of the specific heat of the body.

Principal Forbes' well-known experiments on the conductivity of iron * are the most trustworthy, and will illustrate what has been written and show the student how experiments might be carried out on other metals.

A long bar, I B (in Forbes' experiment, 8 feet by $1\frac{1}{4}$ inch square), fixed on non-conducting or insulating supports, I S, has one end inserted into a pot of melted lead, M L, or solder, kept at a constant temperature by the Argand gas burner, G B. The bar has small holes drilled in it, into which the bulbs of the various accurate thermometers, T_2, T_3 . . . T_{11}, are introduced, a little mercury being poured into the holes so as to form good contact between the bulbs of the thermometers and the bar. The bar is first brought to a uniform temperature, by being left in the laboratory all night without the application of heat. The end is then inserted into the bath of melted lead, and the rise in temperature noted by each of the thermometers, those nearest to the bath beginning to rise first, and then the next, and so on to the last, until finally each of them arrives at a fixed temperature, with a gradual fall between each, graphically represented in the figure by the length of the thermometer stems. The quantity of heat which now passes per minute across any particular transverse section of the bar is constant, and is equal to the product of the cross area, the conductivity, and the fall of temperature at that section. Hence, the quantity of heat passing is expressed by a definite multiple of the unknown conductivity. But that heat does not raise the temperature of the bar beyond the section in question, for the temperature has become stationary, owing to just as much heat passing into the air by cooling as flows into the bar from the leaden bath. To find this rate of cooling, a short bar of the same cross-section and material as the long one, with a thermometer stuck into it, is highly heated and allowed to cool, the rate of cooling being noted by taking frequent readings *at exactly equal intervals of time*—say every half-minute. The heat lost per minute per unit of length, at each temperature, within the range employed, is thus obtained, and a calculation made of what the long bar lost at any particular cross-section.

Principal Forbes found by his experiments that the conductivity of iron for heat, like its conductivity for electricity, diminishes with a rise of temperature. This similar effect on the two forms of energy, heat and electricity, does not appear

* *Trans. Roy. Soc. Edin.*, 1861-2.

however to be common to the other metals experimented upon by Professor Tait; but, as he remarks, "the whole subject, as far as experimental details are concerned, is still in a very crude state."

The value of k in the following expression gives the *thermal conductivity* of a substance at a given temperature in accordance with the definition:—

$$Q = k\,A\,\frac{t_2 - t_1}{x}\,.\,T,$$

Where Q denotes the Quantity of heat that flows in time, T.
A " " Cross area, or the area of each of the opposite faces of the plate.
x " " Thickness of the substance.
t_1, t_2 " " Temperatures on each side of the plate.

From which we see, as has been already remarked, that the quantity of heat which flows by conduction through any substance is directly proportional to the area, and to the difference of temperature between its faces, and inversely proportional to the thickness.

In most experiments the value of the *thermal conductivity* constant, k, is given in accordance with the centimetre, gramme, second, or C.G.S. system of units, and not in the more familiar English foot-pound-minute system. The engineering student will find the following table, taken from the best source—viz., Sir William Thomson's article on "Heat" in the *Encyclopædia Britannica*, 1880 (where the values for the constants, k, c, and $\frac{k}{c}$, are all in C.G.S. units), of considerable interest. From these results, we see that the thermal conductivity of copper is 500 times that of water, and 20,000 times that of air, while iron is 80 times that of water, and 3,500 times that of air. These are important facts to bear in memory, for it shows us that the transmission of heat from the radiant burning coal or charcoal in our furnaces or domestic fire-places on one side of a boiler-plate, kettle,* or frying-pan, to hot water, steam, or melted fat on

* The late Mr. Foulis, M.Inst.C.E., General Manager of The Glasgow Corporation Gas Works, has found, in connection with his numerous experiments on water-heating apparatus for houses and railway carriages worked by gas flames, that thin cast-iron transmits heat more rapidly and effectually to water than copper or other smooth metals of the same thickness and area. This is probably due to the numerous small rough points on the surface of the cast-iron next to the water taking up the heat vibrations and communicating them to the liquid more thoroughly than the much smoother surface of copper or wrought-iron—*A. J.*

DIFFUSIVITIES (THERMAL, MATERIAL, AND ELECTRIC).

Substance.	Thermal Conductivity. k.	Thermal Capacity of Unit Bulk. c.	Diffusivity.* k/c.	Authority.
Copper,	0·91	0·845	1·077	...
Iron,	0·16	0·875	0·185	...
Air, Oxygen, Nitrogen, Carbonic oxide,	0·0000049	0·000307	0·16	Clausius and Maxwell, according to kinetic theory.
Carbonic acid,	0·000038	0·000428	0·088 8	Clausius and Maxwell, according to kinetic theory.
Hydrogen,	0·00034	0·000307	1·12	Clausius and Maxwell, according to kinetic theory.
Underground strata (rough average),	0·005	0·5	0·01	Forbes and W. Thomson.
Wood,	0·0005	0·39	0·001 3	...
Water,	0·002	1·00	0·002 2	J.T. Bottomley.

* What Clerk Maxwell calls Thermometric Conductivity.

the other side, goes on as if the thermal conductivity of the metal were infinite, or, in other words, the resistance to the transmission of heat through the metal, is as nothing compared to the resistance which it meets with from the liquid or gas.

It is important that the engineer should appreciate the relative conducting powers of the different metals that he has to deal with. For instance, the fire-box of a locomotive is made of copper in preference to iron, partly on account of its greater conductivity and partly on account of its withstanding the destructive action of the fire. Mild steel is, however, now being largely used. Has any one yet tried the relative conductivities of the two?* Again, the outside of boilers and cylinders are carefully lagged with some bad conducting substance, so that as little heat as possible may escape therefrom. The following table gives roughly the relative conducting powers of a few of the more common metals:—

Substance.	Relative Conductivity.
Copper,	100
Brass,	30 {and upwards, according to percentage of copper in it.
Zinc,	30
Iron,	16
German Silver,	10
Water,	0·2

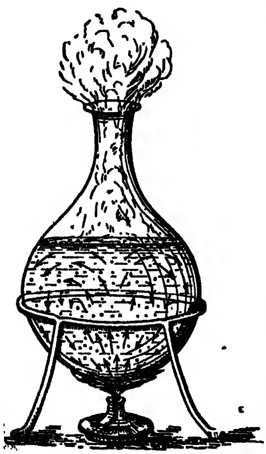
* I am unable to find a good and reliable table of the conducting powers of most of the metals. This subject requires to be taken up and experimented upon.—*A. J.*

As it is frequently of importance to engineers to know the relative conducting powers of bad conductors for purposes of lagging boilers, steam pipes, and cylinders, we extract the following table, taken from *The Proceedings of the Philosophical Society of Glasgow for* 1884, by J. J. Coleman, F.C.S. (the inventor of the well-known Bell-Coleman freezing machine). The experiments are the latest, and were carried out with great care by means of a modification of the Lavoisier Calorimeter:—

RELATIVE CONDUCTING POWERS FOR HEAT.

Silicate cotton,	100	Charcoal,	140
Hair felt,	117	Sawdust,	163
Cotton wool,	122	Gas-works breeze,	230
Sheep's wool,	136	Wood and air space,*	280
Infusorial earth,	136		

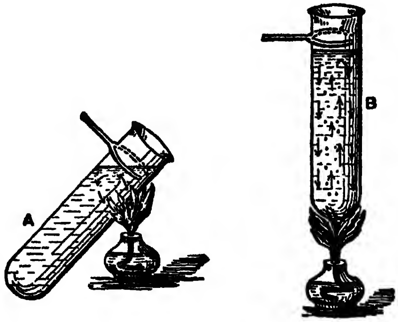
Convection.—When the application of heat to a fluid causes it to expand or to contract, it is thereby rendered rarer or denser than the neighbouring parts of the fluid; and if the fluid is at the same time acted on by gravity, it tends to form an upward or downward current of the heated fluid; this is accompanied with a current from the more remote parts of the fluid in the opposite direction. This action is rendered very apparent by the following simple experiment:—



Take a flask partially filled with water, mix a few grains of bran with it, and apply a lighted spirit-lamp to the bottom of the flask. In a few minutes the water will be seen to circulate in the direction shown by the arrows in figure. The water nearest the flame is rendered lighter, and, therefore, rises upwards, while the denser water falls under the action of gravity, to be in turn heated and raised. The actual transfer of heat throughout the water takes place by conduction, but the diffusion is much assisted by the motion of the fluid, or convection currents, as they are termed.

* Wood and air space, although the best heat conductor in the list, is often used as a non-conductor lagging for boilers, &c., on account of its cheapness and ease of application, but it is not a safe lagging for marine boilers, for it has been known to take fire, *e.g.*, in the s.s. "*John Pender*," one of the Eastern Telegraph Company's cable repairing steamers, in which the author frequently sailed as chief electrician. Charcoal, if only 1½" thick,

The following experiment is also very instructive:—Take a test tube filled with water (left hand, Fig. A), and apply a spirit



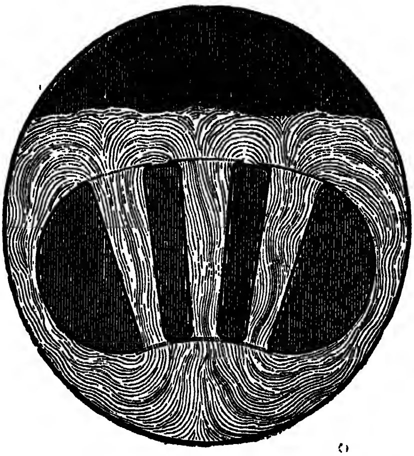
lamp near the surface of the water. You may hold it there for ten minutes or more, and the water at the bottom of the tube is scarcely perceptibly warmer than at first. Now apply the lamp to the bottom of the tube (right hand, Fig. B); in a few minutes the water begins to boil. Why this difference? The convection currents set up, have assisted the naturally bad conducting power of the water by bringing, in turn, every portion of it into close proximity with the source of heat (see Fig., p. 70).

It is for the reasons just mentioned, that the fire-place in a boiler is placed near the bottom instead of near the surface of the water, and it is of great moment not only to give a free and easy path for convection currents in boilers, but to stimulate them by such appliances as hydro-kineters. The better the circulation of the water in a boiler, the more rapidly will it be heated and the steam generated. In many boilers (such as those used on board steamers) the internal construction is so mixed up with tubes and stays, that the water has great difficulty in passing from out-of-the-way corners to the more highly heated parts over the flues; and, if circulation is not assisted, the convection currents "short circuit," as it were (to use an electrical term), and thus leave the more remote portions in comparative chill. For a similar purpose, large boiler flues are provided with "baffling plates," to compel the hot gases

is not suitable for boiler lagging, for in the s.s. "*Volta*," belonging to the same Company, a temperature of about 180° F. was observed on the surface when coated to that depth. This lagging was removed, but it might have done very well if put on thicker, say 3". Leadbetter & Company's self-setting non-conducting composition, which looks very much like soft chalk, is said to do very well, and has this advantage, that it can be laid on while the boiler is cold. Silicate cotton although the best non-conductor or heat insulator in the list, is dear and friable.—*A.J*

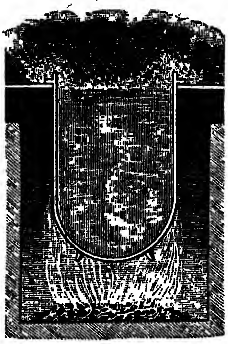
to take a circuitous course, in order that eddies may be formed, and for the further object of promoting a better mixture of air with the inflammable gases.

The art of promoting a good draught in a furnace, or of properly ventilating a building or a ship, depends upon promoting and guiding the convection currents in the proper direction, avoiding sharp bends and contractions. The draught produced by a chimney depends directly upon the difference of the weights of the columns of air (of the cross section and height of chimney), which descend to feed the fires and rise through the chimney. Hence the draught depends upon the height and the cross-sectional area of chimney, or difference of temperature between the gases at the bottom and top of chimney.

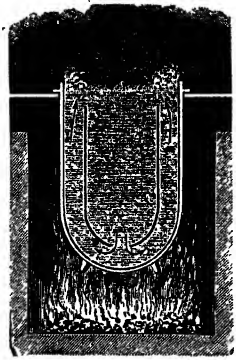


Cross Section through the Combined Oval Flue of the Galloway Boiler, showing the Supposed Circulation of the Water due to the Conical Tubes.

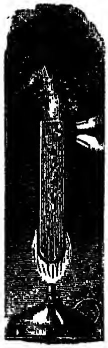
The Ebullition and Circulation of Water in Water-Tube Steam Boilers.—The following eleven figures will give students a very good idea of the circulation of water through the tubes and evolution of a water-tube boiler. These figures are taken from a lecture delivered at Cornell University by Mr. George H. Babcock, with the kind permission of Messrs. Babcock & Wilcox.



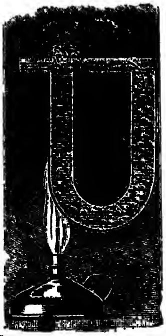
INTERFERENCE OF CONVECTION CURRENTS.



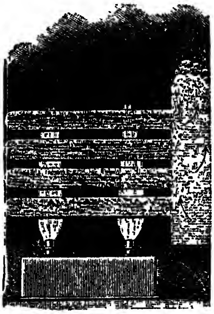
PREVENTING INTERFERENCE BY DIVIDING CONVECTION CURRENTS.



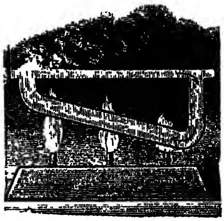
GEYSER-LIKE ACTION.



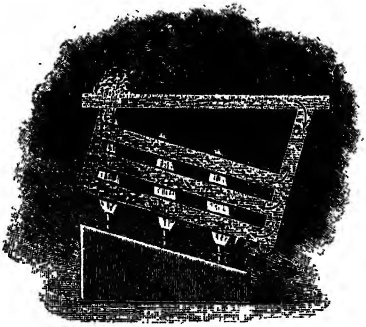
CIRCULATION THROUGH A U-TUBE.



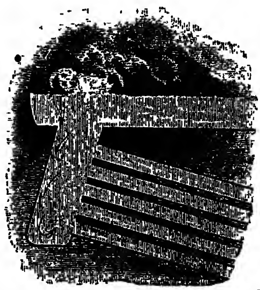
IMPERFECT CIRCULATION WITH HORIZONTAL TUBES.



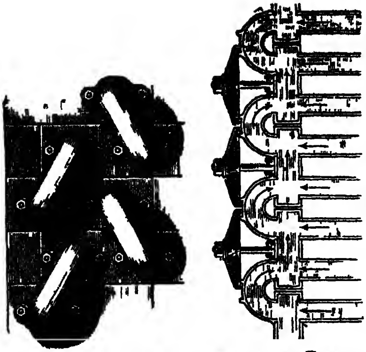
INCLINED HEATING SURFACE CAUSES BETTER CIRCULATION.



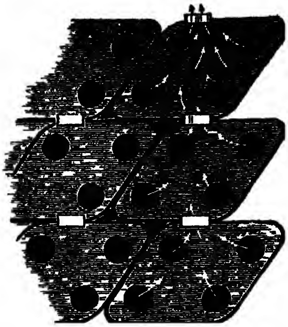
INCREASED AND INCLINED HEATING SURFACE CAUSES STILL BETTER CIRCULATION.



DISADVANTAGES OF LARGE UPTAKE BY PERMITTING CONTRA-DOWN CURRENTS.



DISADVANTAGES OF PLACING RETURN BENDS OPPOSITE THE ENDS OF THE TUBES.



DISADVANTAGES OF DIFFERENT CROSS SECTIONS IN THE FLOW OF CURRENTS.

Now, after studying these eleven figures, the student should refer to the folding-plate and sections of the Babcock-Wilcox Boilers in Lecture XXVIII., where he will see the stage at present reached by the foregoing evolution or development of circulating water in tubular boilers.

LECTURE V.—QUESTIONS.

1. What is meant by capacity for heat or thermal capacity? The specific heat of mercury being ·033, how much, at the temperature of 240° F., will be sufficient to raise 12 lbs. of water from 50° to 58° F.? *Ans.* 15·98.

2. What will be the relative capacities for heat of the same volumes of air, carbonic oxide, steam, and hydrogen at the same pressures if their densities are as 14·4, 14, 7, and 1 respectively? (Prove answer by arithmetic.) *Ans.* All equal, because the capacity for heat of equal volumes is inversely as the density.

3. What do you mean by conduction and convection, as applied to heat?

4. Describe an experiment by which you would show that water is an extremely bad conductor of heat. For what reason should heat be applied from below when it is required to heat a large mass of water rapidly?

5. What is the object of facilitating the circulation of water in boilers? State and illustrate two ways by which the circulation of the water in a boiler increases the efficiency or ratio of heat in steam to heat applied to heating surfaces

6. What is the effect on the circulation of the water by having horizontal tubes stopped at one end, or return bends opposite the tubes in the water-tube boiler? Also, why is it necessary to guard against having the uptake too large at the upper end of the tubes in a water-tube boiler?

7. Trace the evolution of the water-tube boiler by neatly-drawn sketches and concise descriptions of each.

8. When draught is produced by a chimney, upon what things does the magnitude of the draught depend? In order to approximate to the temperature of the gases at the base of a chimney, a mass of iron weighing 8 lbs was placed in them, and after remaining a considerable time was removed and submerged in 100 lbs. of water at 50° F., when it was found that the temperature of the water was raised to 55° F. Find the temperature of the gases, having given that the specific heat of iron is one-ninth. (C. & G., 1903, O., Sect. C.)

LECTURE VI.

CONTENTS.—Nature of Heat—Heat is not a Substance—Rumford, Davy, and Joule's Experiments—Conversion of Work into Heat—First Law of Thermo-dynamics—Joule's Mechanical Equivalent of Heat—Latest Equivalents for the B.T.U.—Questions.

UNTIL the end of last century, two rival theories had been entertained regarding the nature of heat. One, that heat consisted of a subtle elastic fluid, termed caloric, penetrating through the pores or interstices of matter, like water in a sponge; the other, that it was an internal commotion among the particles or molecules of matter.

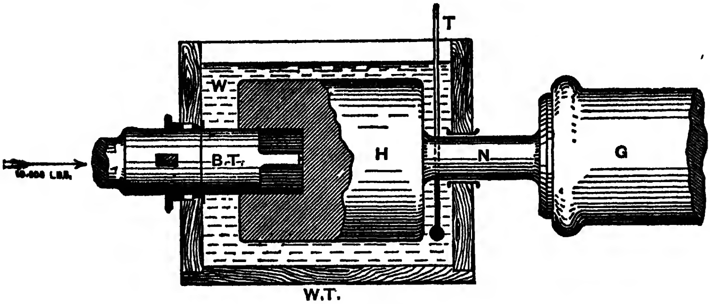
The former of these theories, or hypotheses, that heat is matter, called the "materialistic doctrine of heat," taught by Professor Black of Glasgow University and others, was most conclusively overthrown by the celebrated experiments of Count Rumford and Davy. It is very remarkable, however, that fifty years elapsed before scientific men generally became converted to the conclusions to be drawn from them. It was not until Joule, during the period extending from 1840 to 1849, had supplied several fresh proofs that heat is not a material substance, but one form of energy, which may be applied to, or taken from bodies in various ways, and that the amount of energy, in whatever form applied or removed, may be estimated in mechanical units of work or foot-pounds, that what is now known as the *Kinetic theory of heat*, became generally accepted, and the science of thermo-dynamics placed on a firm basis.

Count Rumford's experiments on the production of heat by friction, were carried out in the following manner, and communicated to the Royal Society in 1798:—

In casting guns it was usual to leave a projecting cylindrical "head" of metal at the muzzle, so as to insure sound metal in the gun. The guns were cast in a vertical position with the muzzle end upwards, very much in the same way as large water or gas pipes are now made. The effect of adding the "head" to the casting, being to add pressure to the fluid metal in the lower parts, thus expelling air and gases towards the surface, and into the "head," which was cut off before boring out the gun.

Rumford obtained a casting for a six-pounder brass gun from the military arsenal at Munich, and surrounded the "head,"

H, by a wooden trough, W T, containing about 18 lbs. of water, W, at 60° Fah. The machinery which rotated the gun, G, was driven by two powerful horses. A blunt boring tool, B T, which was made of steel, 3·5 inches diameter, was forced



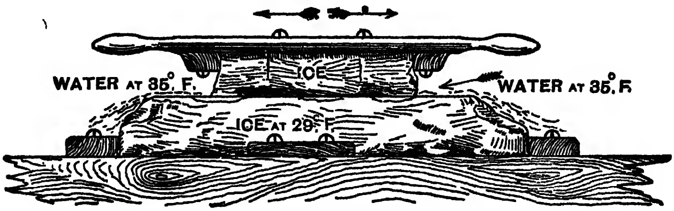
COUNT RUMFORD'S EXPERIMENT.

G	for Gun.	W T	for Wooden trough.
N	„ Neck.	W	„ Water.
H	„ Head.	T	„ Thermometer.
B T	„ Boring tool.		

against the head, H. This boring tool was held firmly in a rest, and pressed forward by means of a screw with an estimated pressure of 10,000 lbs. The result of this experiment was that, the heat generated by the friction between the blunt boring tool and the metal of the head, was partly conducted through the neck connecting the head with the gun, and partly absorbed by the water in the trough, so that the temperature of the water rose at the end of an hour to 107° F., in an hour and a-half to 142° F., in two hours to 178° F., and, finally, at the end of two and a-half hours the water boiled. Count Rumford said—"It would be difficult to describe the surprise and astonishment expressed in the countenances of the by-standers on seeing so large a quantity of water heated, and actually made to boil without any fire!" He adds—"By meditating on the results of these experiments, we are naturally brought to that great question which has so often been the subject of speculation, namely—What is heat? Is there any such thing as an igneous fluid? Is there anything that, with propriety, can be called caloric?" And, further—"It is hardly necessary to add that anything which an insulated body or system of bodies can continue to furnish without limitation, cannot possibly be a material substance; and it appears to me to be extremely difficult, if not impossible, to

form any distinct idea of anything capable of being excited, and communicated in the manner heat was excited, and communicated in these experiments except it be motion."

Davy's experiment on the melting of ice by friction, announced by him in 1799, in his first published work, entitled—*An Essay on Heat, Light, and Combinations of Light*, was regarded at the time as a complete refutation of the materialistic doctrine of heat.



SIR HUMPHREY DAVY'S EXPERIMENT.

In an atmosphere at a temperature of 29° F., he rubbed together two small slabs of ice with the result (as shown in the fig.) that the ice was melted at the surfaces of contact, producing water at a temperature of 35° F. Now, as we saw in Lecture IV., a mass of water contains an absolute quantity of heat greater than an equal mass of ice, and it is, therefore, impossible to account for the presence of the increased temperature on the assumption that heat is a material substance. Davy said—"The immediate cause of the phenomenon of heat is motion, and the laws of its communication are precisely the same as the communication of the laws of motion."

Maxwell, in his *Theory of Heat*, p. 306, says—"The molecules of all bodies are in a state of continual agitation. The hotter the body is, the more violently are its molecules agitated."

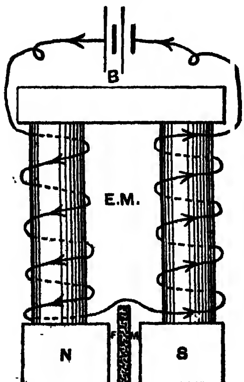
Joule's experiments, carried out between 1840 and 1849, recalled the attention of scientists to Rumford and Davy's doctrine regarding the nature of heat, and gave us the means of estimating with exactness the quantity of work required to generate a certain quantity of heat.

We shall describe only two of Dr. Joule's famous experiments, and for a complete list of them we refer the student to Sir William Thomson's article on "Heat," in *The Encyclopædia Britannica*, 1880, p. 34.

Dr. Joule filled a copper tube with a fusible metal or alloy, F M (such as that used by the printers in making stereotype castings of types), which fuses at a low temperature, and

revolved the tube rapidly between the poles, N, S, of a strong electro-magnet, E M. The result of this was that the temperature of the alloy rose in a few minutes to the melting point, and the alloy could be poured from the copper tube. What agency was at work to fuse the metal? There was no friction between the revolving tube and any other part of the mechanism, for the tube rotated quite clear of the poles in the space between them; neither was it due to any friction from the spindle carrying the copper tube, for, if the battery or dynamo was disconnected (and thus no magnetism evoked) the tube might be revolved at the same speed as before, without any observable rise in temperature in the alloy. One circumstance was, however, made very apparent, viz., that it required much less effort to revolve the tube in the latter case than in the former, and herein lies the key to the whole secret.* A certain proportion of the power devoted to revolving the tube between the magnetised poles is expended in creating electric currents in the copper tube, and in the metal contained therein. These currents agitate and vibrate the molecules of the metal so very rapidly amongst themselves, that heat results from the forces at work overcoming the inter-molecular friction.

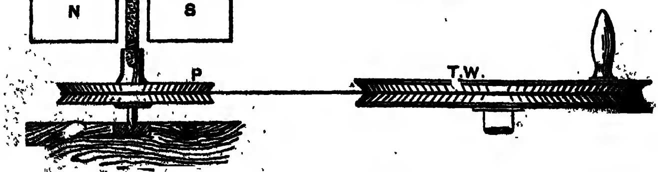
To prove that electric currents are so generated, we have only to cite the case of the now well-known Dynamo, where the copper



* This experiment is shown to my class by means of the small dynamo belonging to the College of Science and Arts driven by one or two students. The difference in the effort required by them to drive the dynamo, or to revolve the metal tube in the two cases, is thus brought home to them in a manner quite unattainable by any mere description or diagram.

INDEX.

B for Battery or dynamo
E M „ Electro-magnets.
N, „ North and south poles.
F M „ Fusible metal in a copper tube
P „ Pulley.
T W „ Turn wheel.



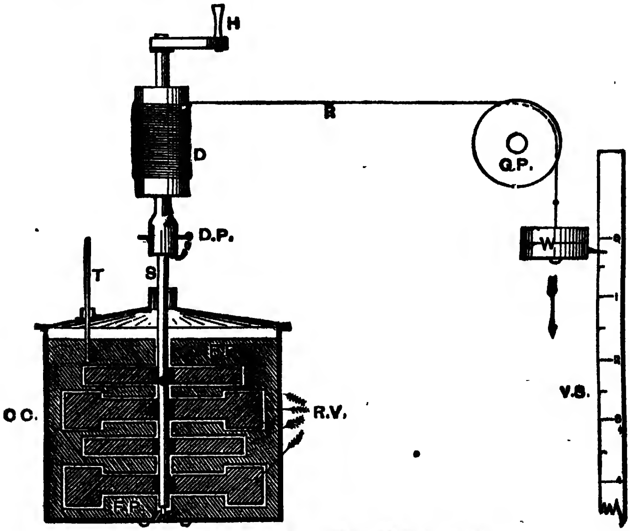
JOULE'S MAGNETO-ELECTRIC EXPERIMENT

wires forming the armature, when revolved between the powerful magnetic poles, have strong currents excited in them; and to show that such currents are capable of producing heat, we have only to pass them through a thin metal wire or an incandescent lamp, with the result that the wire is heated to a white heat, or even fused, and the carbon filament made to glow with a brilliant incandescence.

We have in this experiment of Dr. Joule's a beautiful example of the double conversion of energy, viz., (1) mechanical energy into electrical energy, and (2) electrical energy into heat energy.

Joule's favourite experiment was the conversion of work into heat by the stirring of water. He arranged his apparatus in a manner similar to that shown in the figure.

A known weight, W, was allowed to fall through a known height, and in doing so to revolve vanes or paddles, R V, inside



JOULE'S WATER-STIRRING EXPERIMENT.

V S for Vertical scale in feet.
W ,, Weight.
R ,, Rope or twine.
G P ,, Guide pulley.
D ,, Drum.
H ,, Handle.
S ,, Spindle.

O C for Copper cylinder.
T ,, Thermometer.
R V ,, Revolving vanes or paddles (8 sets).
F P ,, Fixed plates (4 sets).
D P ,, Disconnecting pin.

a copper cylinder, C C, containing a known weight of water; thus churning the water against the fixed plates or stationary screens, F P. The effect of this churning was to raise the temperature of the water, by imparting to it a certain quantity of heat, depending on the product of the weight into the space through which it fell, or the foot-pounds of work expended. We need not enter into the many details of Dr. Joule's carefully conducted experiments, whereby he eliminated from his results the effect of friction in the guide pulley, G P, as well as the effects of radiation and conduction of heat to or from the apparatus during the time of the experiment, &c. It will suffice to give his final result and an example.

The British Association in 1870 requested Joule to reinvestigate the subject for the purpose of giving greater accuracy to the determinations by his fluid friction method, with the final result of proving that 772·43 *foot-pounds* (*at the latitude of Manchester*) *are equal to the quantity of heat required to warm from* 60° *to* 61° *Fah. a pound of water weighed in vacuum.* This has been termed "Joule's Mechanical Equivalent of Heat," or, shortly, "Joule's Equivalent," and is denoted by the letter, J, and in round numbers we say, 1 *British thermal unit* = 772 *ft.-lbs.* Reduced to the centimetre gramme second or (C.G.S.) system, it is equivalent to about 42 million "ergs" or units of work for one gramme of water raised in temperature from 0° to 1° C.

For instance, suppose that, with Joule's apparatus we had a weight of 77·2 lbs., and allowed it to fall through a height of 10 feet, and in doing so, the mechanical work (772 ft.-lbs) would be converted into heat by churning 1 lb. of water at 60° F., we should find (if all extraneous losses were avoided) that the water had risen in temperature to 61° F., when the weight passed the 10th foot; or, if we take 1 lb. of water at 60° F., and raise its temperature 1° F., by any method whatever, the quantity of heat imparted to it (viz., 1 thermal unit), if converted into mechanical energy by a perfect heat engine, would perform 772 ft.-lbs. of work, or raise 772 lbs. 1 foot (see foot-note to next page).

First Law of Thermo-dynamics.—*Heat and work are mutually convertible, and Joule's equivalent is the rate of exchange.*

The importance of this mutual relation between *heat* and *work*, cannot be too strongly impressed on the student at the very outset of his studying steam and the steam engine. In this lecture it has been shown, that the expenditure of so many *units of work* produces under the circumstances noted, an exact and unvarying equivalent of so many *units of heat;* and we shall see in future lectures, how the expenditure of so many *units of heat* produces an equivalent in *units of work.*

A familiar illustration of the foregoing principle of the mutual convertibility of heat and work is that of the Locomotive Engine. In the furnace

we have the production of heat by the combustion of coal. A proportion of this heat is imparted to the water in the boiler thus raising steam. The steam on being admitted to the cylinders parts with a portion of its heat in the act of doing the work of propelling the pistons, and thus moving the train. Again, when the train is nearing a station the steam is shut off, and the brakes applied. Then the stored work is converted into heat, which may be observed by sparks issuing at the brakes and by feeling the increased temperature of the brakes, wheels, and rails.

Example I.—Suppose a locomotive burns 6 lbs. of coal per horse-power hour, and that every pound of coal burned in the furnace gives up to the water in the boiler 10,000 British units of heat, we have—

$$6 \text{ lbs.} \times 10{,}000\,u = 60{,}000 \text{ units of heat per H.P. hour.}$$

But— $$1 \text{ H.P.} = 33{,}000 \text{ ft. lbs. per minute, or}$$
$$= 33{,}000 \times 60' = 1;980{,}000 \text{ ft.-lbs. per hour}$$

And— $$772 \text{ ft.-lbs.} = 1 \text{ unit of heat.*}$$

$$\therefore \frac{1;980{,}000}{772} = 2567{\cdot}2 \text{ units of heat converted into work every hour.}$$

Consequently— $$60{,}000\,u : 2567{\cdot}2\,u :: 100 : x = 4{\cdot}27,$$

Or the locomotive only converts 4·27 per cent. of the total heat generated in the furnace into its equivalent of work in the cylinder.

Example II.—Suppose that the energy of the train when the brakes are put on is equal to 16; 500,000 ft.-lbs.

Then, 16; 500,000 ÷ 772 = 21,373 units of heat, or an amount of heat is generated at the brakes, wheels, and rails, &c., which would raise 213·73 lbs. of water 100° F.

Note.—We thus see from these two examples that the transformation from work into heat is more easy and complete than from heat into work.

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* Prof. Reynolds, M.Inst.C.E., F.R.S., in his recent and careful experiments with "Froude's Water Dynamometer" (see the author's *Text-Book on Applied Mechanics and Mechanical Engineering*, vol. i., 4th and later editions) at the Engineering Laboratory, Owens College, Manchester, found that 777 ft.-lbs. of work were equivalent to *one British Thermal Unit*. Profs. Rowland, Griffith, and Schuster have, however, ascertained that the mean of their and other experiments gave the number 778 ft.-lbs. = 1 B.T.U.

I am loth to change all the 772 values in this book, until an "International Congress of Engineers" authorise a standard value for this important quantity.

Lecture VI.—Questions.

1. Give free-hand sketches with index of parts, and a description in *your own* words of Rumford's, Davy's, and Joule's experiments.

2. State in your own words what you consider heat to be, and give Joule's mechanical equivalent for one British thermal unit.

3. How has the work done in raising the temperature of a pound of water through one degree been ascertained? A pound of coal gives out during combustion, 12,000 units of heat; how much work in foot-pounds could be done per pound of coal burned, if there were no waste? *Ans.* 9,264,000 ft.-lbs.

4. It is estimated that every pound of average steaming coal burned in the furnace of a boiler gives out 13,000 units of heat. It is found that a good compound engine and boiler requires 2 lbs. of coal per hour per indicated horse-power. What is the efficiency of the combined boiler and engine? *Ans.* 9·86 per cent.

5. Give another illustration of the first law of thermo-dynamics than that in the lecture, and work out an arithmetical example, and thus show that the transformation from mechanical work into heat is much more complete and efficient than from heat into work.

6. Define a unit of heat. A steam engine indicates 25 H.P., how many units of heat does it convert into useful work per minute? *Ans.* 1,068·65.

7. The following data are obtained during a gas engine trial:—I.H.P. = 42·6. Gas used per hour = 815 cubic feet. Cooling water used per hour = 320 gallons. Inlet temperature of jacket water = 61·5° F. Outlet temperature of jacket water = 125·7° F. Calorific value of 1 cubic foot of gas = 635 B.T.U. Make out as far as you can a heat account for this engine. (C. & G., 1900, H., Sec. B.)

8. A man, working 8 hours a day for 300 days in the year, does work at the rate of one-tenth of a horse-power. One lb. of coal, having a calorific value of 15,000 thermal units, is burnt in a boiler having an efficiency of 70 per cent., which supplies steam to an engine having an efficiency of 15 per cent. Find how many years the man will have to work in order to give out as much useful work as 1 ton of coal used in the above plant. (C. & G., 1902, O., Sec. C.)

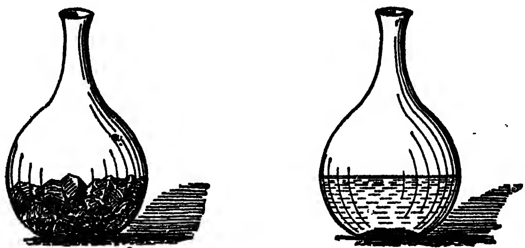
9. An oil engine of 2½ horse-power drives a motor car at a speed of 15 miles an hour. If the efficiency of the engine is 10 per cent., and the calorific value of the oil 20,000 thermal units, how much oil is consumed in a run of 100 miles? (C. & G., 1903, O., Sect. C.)

LECTURE VII.

CONTENTS.—Sensible and Latent Heats of Water and Steam—Temperature and Pressure of Steam—Regnault's Experiments—Tables I. and II. on Properties of Steam—Explanations of Sensible and Latent Heats, &c. —Mercurial Pressure and Vacuum Gauges—Bourdon's Pressure and Vacuum Gauges—Schäffer's Pressure Gauge and Thalpotasimeter—Questions.

Sensible and Latent Heats of Water and Steam.—Hitherto we have dealt with heat when imparted to or abstracted from bodies as indicated by a rise or fall of temperature in the body. It has been customary to call this condition *sensible heat;* but there are exceptional cases in which temperature does not vary in a mass of matter when heat is communicated to it, from, or taken from it, to, external matter. For instance, when the body is ice at the melting point, heat communicated to it does not raise its temperature above 32° F., or, if the body be water at the boiling point in the open air, heat slowly communicated to it, in however great a quantity, does not raise its temperature above 212° F., at the normal pressure of the atmosphere. This heat is termed *latent heat.*

A short account of Professor Black's well-known experiments carried out about 1762, will serve to illustrate the difference between what is termed the "sensible" and the "latent" heat of a substance.



BLACK'S EXPERIMENT ON LATENT HEAT OF WATER.

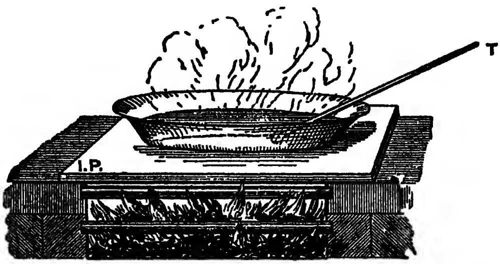
Black procured two glass flasks, in one of which he placed 5 ozs. of ice at 32° F.,* and in the other 5 ozs. of water at the same

* The ice was beginning to melt, and his estimate of the temperature at the surface was 33° F.

temperature. He suspended them within a short distance of each other in a room which remained at a uniform temperature of about 47° F. He observed that in *one* half-hour the water increased in temperature by 7° F., but that it took *twenty* half-hours for the whole of the ice in the other flask just to become melted, and he reasoned thus—that from the time required the amount of heat which had entered the ice must have been *twenty* times as much as that which entered the water. He, therefore, computed that the *latent heat* of water must be 7×20 (half-hours) $= 140$.

Another experiment of Black's was that of placing a lump of ice in an equal weight of water at 176° F., with the result that when the whole of the ice had melted, the temperature was no greater than that of water just ready to freeze. Therefore, assuming the final temperature of the mixture to have been 33° F., we have $176 - 33 = 143$, as the amount of heat required to melt the ice, or *the latent heat of water*.

In this estimate he was very near the truth; for, even at the present day the mean results of some of the best experimenters appears to be, that 143 British thermal units of heat are absorbed, or become latent, in the conversion of 1 lb. of ice into water at the same temperature; and, consequently 143 B.T.U., are given out or let free in the conversion of 1 lb. of water at 32° F., into ice at the same temperature.*



BLACK'S EXPERIMENT ON THE LATENT HEAT OF STEAM.

Black's third experiment consisted in placing a flat tin dish on a hot plate over a fire; into this plate he put a small quantity of water at 50° F., and observed that after 4 minutes the water

* The latent heat of water by the Centigrade scale is 79·4 for $\frac{143 \times 5}{9} = 79{\cdot}4$, say 79 units of heat required to convert 1 lb. of ice at 0° C., into 1 lb. of water at the same temperature.

began to boil, and in 20 minutes more it had all evaporated. Now, since the water increased by $(212° - 50°) = 162°$ in 4 minutes, he reasoned that it must have been receiving heat at the same rate throughout the experiment, or that, in 20 minutes it had absorbed five times as much as in the first 4 minutes without any apparent rise in temperature as indicated by the thermometer, or, $5 \times 162 = 810$—Black's estimate of the latent heat of steam.

In this last estimate Black was incorrect, as might be expected, from the rough nature of his experiment. It has since been found that the *latent heat of steam* at atmospheric pressure is 966·6. In other words, it requires 966·6 British thermal units of heat to convert 1 lb. of water at 212° F., into steam at the same temperature, or 1 lb. of steam at 212° F., gives out 966·6 B.T.U., in being condensed into water at the same temperature.*

The following definition of sensible and latent heat will now be quite clear:—

"Heat given to a substance, and warming it, is said to be *sensible* in the substance. Heat given to a substance, and *not* warming it, is said to become *latent*" (*Sir Wm. Thomson*).

Latent heat is the quantity of heat which must be communicated to unit mass of† a body in a given state, in order to convert it into another state without changing its temperature (*Maxwell*).

Temperature and Pressure of Steam.—When water is confined in a closed vessel, and heated, the pressure of the vapour contained therein continually increases. The precise temperature which corresponds to any particular pressure, has been made the subject of very careful inquiry by Regnault and others. Before quoting Regnault's results, we shall illustrate these phenomena by means of a simple apparatus, termed Marcet's boiler.

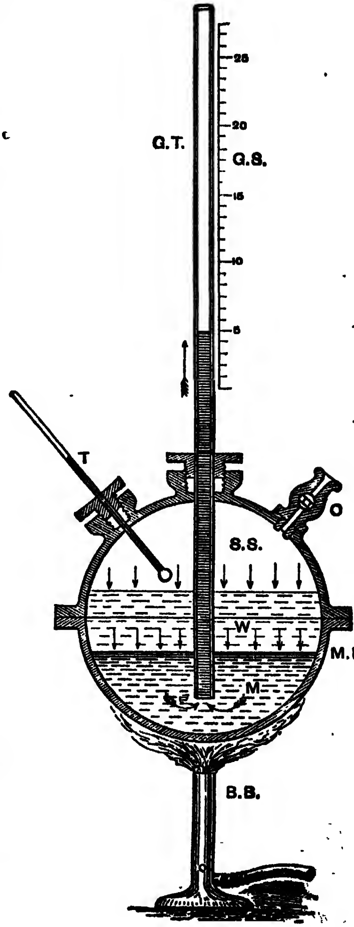
On applying heat from the Bunsen burner, B B, steam is generated from the water, W, and the temperature as it rises is noted by the thermometer, T. Simultaneously the column of mercury rises in the tube, and the height from the free surface of the mercury may be read off (roughly) on the graduated scale, G S. When the temperature has arrived at 233° F., the mercury will be observed to have risen about 15 inches, corresponding to

* The Latent Heat of Steam by the Centigrade scale, is, therefore, $\frac{966·6 \times 5}{9} = 537$; or, 537 times the quantity of heat absorbed in raising 1 lb. of water by 1° C.

† I have added the words (unit mass of) to Maxwell's definition, because it appears deficient without them. When we speak of 143 as the latent heat of water, and 966 as the latent heat of steam, it is understood that 143 and 966 units of heat are required respectively for every 1 lb. (or unit of mass) to change the state from solid to liquid, and from liquid to gaseous.—*A. J.*

a pressure of 7·4 lbs. (half an atmosphere), or 22 lbs. absolute; and when the temperature arrives at 250° the mercury will have risen to about 30 inches, corresponding to a pressure of 14·7 lbs. on the square inch (1 atmosphere), or 29·4 lbs. absolute (*i.e.*, from zero pressure, or what would correspond to a perfect vacuum).

If our glass tube had been longer, and the supply of mercury in the bottom of the boiler sufficient, we might have gone on applying heat and registering still higher pressures with their corresponding temperatures, but the limited experiment has been sufficient to show roughly, that a rise in temperature cannot take place without a corresponding rise in pressure. Mercurial gauges, such as that in the Marcet's boiler, were much used to register the pressure of steam in steam boilers, before the introduction of the Bourdon gauge. (See our Elementary Manual, Lecture XI.)



INDEX.

B B for Bunsen burner.
M B ,, Marcet's boiler.
M ,, Mercury.
W ,, Water.
S S ,, Steam space.
T ,, Thermometer in S S.
G T ,, Glass tube, about 35 in. long.
G S ,, Graduated scale.
C ,, Cock.

Regnault's Experiments.—Our knowledge of the properties of steam is chiefly derived from experiments made by Regnault at the Paris observatory for the French Government in 1847. They

were conducted with the greatest care, and involved immense labour. It is not necessary here to enter into any minute detail of the apparatus he used, but, generally speaking, it consisted of a boiler containing, when half full, about 33 gallons of water, a condenser of suitable dimensions to condense the steam as fast as it was formed, and an air chamber three times the size of the boiler provided with force pumps by means of which any desired pressure could be produced at pleasure. Pressures were measured by means of a column of mercury open to the atmosphere—an arrangement admitting of greater accuracy than any other method, but involving the manipulation of a column of mercury some 50 feet in height, when registering the very high pressures to which he went, viz., over 400 lbs. on the square inch. The air chamber and condenser enabled any desired pressure to be maintained for any length of time. For his more accurate measurements of temperature he used an air thermometer.

Numerous formulæ have been devised for connecting algebraically the relation subsisting between the temperature and the pressure of *saturated steam*. Many of them are defective in as far as they only apply to a limited range,* of which the following is one of the best of these approximate formulæ, as it is nearly correct for *absolute* pressures between 6 and 60 lbs., and it may be also used for pressures between 60 and 120 lbs., by adding 1 to the results.

$$p = \left(\frac{t + 40}{147}\right)^5$$

When p stands for the absolute pressure in lbs. per sq. in.
t „ „ temp. of the boiling point in degrees F.

$$\text{Therefore,} \qquad t = 147\sqrt[5]{p} - 40$$

Which is easily worked out by logarithms for any particular case, but the following, as given by Prof. Rankine, is best—

$$\log. p = A - \frac{B}{\tau} - \frac{C}{\tau^2}$$

Where A, B, and C are constants, and τ, the absolute temperature of the boiling point $= t + 460$ (see Lecture XIII.)

The inverse formula for finding, τ, when you know p, is

$$\tau = 1 \div \left\{\sqrt{\left(\frac{A - \log. p}{C} + \frac{B^2}{4C^2}\right)} - \frac{B}{2C}\right\}$$

* Rankine's formulæ, which represents Regnault's experimental results, very closely refers to the *absolute temperature*, or −460 F. (see Lecture XIII.)

Where

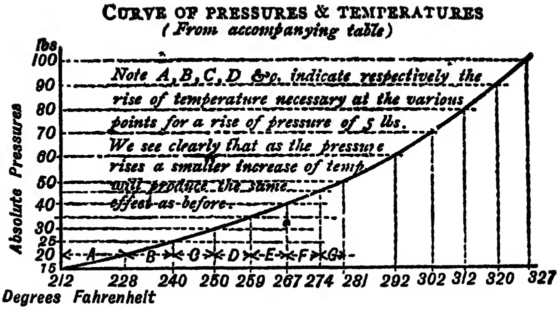
A	Log. B	Log. C	$\frac{B}{2C}$	$\frac{B^2}{4C^2}$
6·1007	3·43642	5·59873	0·003441	0·00001184

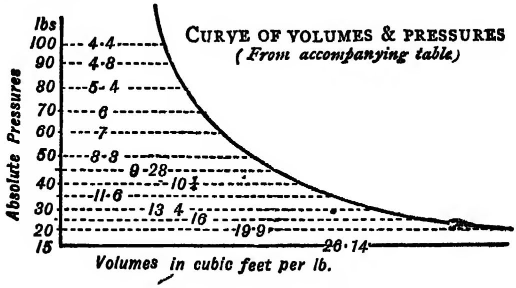
Students should plot out a curve from the following table, representing graphically the relation between pressure and temperature. To do so, set off a scale of temperatures on a horizontal line, and on a vertical line starting from the same point plot the corresponding pressures *to the same scale.* Draw vertical lines from each of the former points, and horizontal lines from each of the latter. Connecting the points of intersection, we have a curve, which shows at a glance how the pressures increase more rapidly than the temperatures. See next page.

TABLE I.—PROPERTIES OF SATURATED STEAM FROM 32° TO 212° F.*

Temperature.	Pressure.		Temperature.	Pressure.	
	Inches of Mercury.	Lbs. per Square Inch, Absolute.		Inches of Mercury.	Lbs. per Square Inch, Absolute.
Fahrenheit	Inches.	Lbs.	Fahrenheit	Inches.	Lbs.
32°	·181	·089	120°	3·430	1·685
35	·204	·100	125	3·933	1·932
40	·248	·122	130	4·509	2·215
45	·299	·147	135	5·174	2·542
50	·362	·178	140	5·860	2·879
55	·426	·214	145	6·662	3·273
60	·517	·254	150	7·548	3·708
65	·619	·304	155	8·535	4·193
70	·733	·360	160	9·630	4·731
75	·869	·427	165	10·843	5·327
80	1·024	·503	170	12·183	5·985
85	1·205	·592	175	13·654	6·708
90	1·410	·693	180	15·291	7·511
95	1·647	·809	185	17·044	8·375
100	1·917	·942	190	19·001	9·335
105	2·229	1·095	195	21·139	10·385
110	2·579	1·267	200	23·461	11·526
115	2·976	1·462	205	25·994	12·770
			210	28·753	14·126
			212	29·922	14·700

* From D. K. Clark's *Rules, Tables, and Data:* Blackie & Son.





REMARKS ON TABLE II.

Saturated Steam is steam in contact with the water from which it is generated. Its physical condition is such, that it is ready on the smallest increase of pressure, or decrease of temperature, to yield some portion as liquid. For a given pressure there is one temperature and one density.

Absolute Pressures are pressures reckoned from a perfect vacuum as zero. Ordinary steam pressure, as measured by steam gauges, is converted into absolute pressure by adding 15 lbs. *N.B.*—In all questions relative to the expansion of steam (Boyle's law, &c.) absolute pressures are to be used.

Temperature.—The second column gives the temperature at which water boils under the given pressure, and the temperature of the steam produced. It also gives (nearly) the units of heat required to raise 1 lb. of water from 32° to boiling point under this pressure *when 32 is subtracted.*

Example.—1 lb. water at 120° raised to boiling point under 50 lbs. pressure. The units of heat required = 281 − 120 = 161 units.

Total Heat, or sum of Sensible and Latent Heat.—This was believed by Watt to be a constant quantity, but elaborate and careful experiments by Regnault have shown that it increases along with the temperature. The formula used in the table for any temperature, t, is:

$$\text{Total Heat, or } H = 1{,}082{\cdot}4 + {\cdot}305\,t^{\circ}.$$

Latent Heat gets less at higher temperatures and pressures.

Relative Volume is the volume of steam generated under a given pressure compared with the volume of the water from which it is produced.

Pressure in Lbs. per Sq. In. above an Abs. Vacuum.	Temperature in Degrees Fahrenheit.	Total Heat in Heat Units from Water at 32°.	Heat in Liquid from 32° in Units.	Heat of Vaporisation, or Latent Heat in Heat Units.	Density or Weight of Cubic Foot in Lbs.	Volume of 1 Lb. in Cubic Feet.	Special Factor of Equivalent Evaporation at and from 212° F.	Heat Equivalent of the External Work done during Evaporation in B.T. Units.	Pressure in Lbs. per Sq. In. above an Abs. Vacuum.
p	$t°$	H	S	L	w	V_S	E	H_E*	p
1	101·99	1113·1	70·0	1043·0	0·00299	334·5	0·9661	61·90	1
2	126·27	1120·5	94·4	1026·1	0·00576	173·6	0·9738	64·23	2
3	141·62	1125·1	109 8	1015·3	0·00844	118·5	0·9786	66·56	3
4	153·09	1128·6	121·4	1007 2	0·01107	90·33	0·9822	66·86	4
5	162·34	1131·5	130·7	1000·8	0·01366	73·21	0·9852	67·74	5
6	170·14	1133·8	138·6	995·2	0·01622	61·65	0·9876	68·44	6
7	176·90	1135·9	145·4	990·5	0·01874	53·39	0·9897	69·11	7
8	182·92	1137·7	151·5	986·2	0·02125	47·06	0·9916	69·66	8
9	188·33	1139·4	156·9	982·5	0·02374	42·12	0·9934	70·15	9
10	193·25	1140·9	161·9	979·0	0·02621	38·15	0·9949	70·57	10
†14·7	212·00	1146·0	180·0	966·0	0·03800	26·36	1·0000	72 00	†14·7
15	213·03	1146·9	181·8	965·1	0·03826	26·14	1·0003	72 51	15
20	227·95	1151·5	196·9	954·6	0·05023	19·91	1·0051	73·67	20
25	240·04	1155 1	209·1	946·0	0·06199	16·13	1·0099	74·57	25
30	250·27	1158·3	219·4	938·9	0 07360	13·59	1·0129	75·41	30
35	259·19	1161·0	228·4	932·6	0·08508	11·75	1·0157	75·90	35
40	267·13	1163·4	236·4	927·0	0·09644	10·37	1·0182	76·70	40
45	274·29	1165·6	243·6	922·0	0·1077	9·285	1·0205	77·20	45
50	280·85	1167·6	250·2	917·4	0·1188	8·418	1·0225	77·76	50
55	286·89	1169·4	256·3	913·1	0·1299	7·698	1·0245	78·21	55
60	292·51	1171·2	261·9	909·3	0·1409	7·097	1·0263	78·65	60
65	297·77	1172·7	267·2	905·5	0·1519	6·583	1·0280	79·02	65
70	302·71	1174·3	272·2	902·1	0 1628	6·143	1·0295	79·39	70
75	307·38	1175·7	276·9	898·8	0·1736	5·760	1·0309	79·75	75
80	311·80	1177·0	281·4	895·6	0·1843	5·426	1·0323	80·12	80
85	316·02	1178·3	285·8	892·5	0·1951	5·126	1·0337	80·39	85

p	t	H	S	L	w	V_S	E	H_E*	p
90	320·04	1179·6	290·0	889·6	0·2058	4·859	1·0350	80·67	90
95	323·89	1180·7	294·0	886·7	0·2165	4·619	1·0362	80·95	95
100	327·58	1181·9	297·9	884·0	0·2271	4·403	1·0374	81·17	100
105	331·13	1182·9	301·6	881·3	0·2378	4·205	1·0385	81·40	105
110	334·56	1184·0	305·2	878·8	0·2484	4·026	1·0396	81·64	110
115	337·86	1185·0	308·7	876·3	0·2589	3·862	1·0406	81·87	115
120	341·05	1186·0	312·0	874·0	0·2695	3·711	1·0416	82·08	120
125	344·13	1186·9	315·2	871·7	0·2800	3·571	1·0426	82·24	125
130	347·12	1187·8	318·4	869·4	0·2904	3·444	1·0435	82·47	130
140	352 85	1189·5	324·4	865·1	0·3113	3·212	1·0453	82·81	140
150	358·26	1191·2	330·0	861·2	0·3321	3·011	1·0470	83·16	150
160	363·40	1192·8	335·4	857·4	0·3530	2·833	1·0486	83·43	160
170	368·29	1194·3	340·5	853·8	0·3737	2·676	1·0502	83·70	170
180	372·97	1195·7	345·4	850·3	0·3945	2·535	1·0517	84·00	180
190	377·44	1197·1	350·1	847·0	0·4153	2·408	1·0531	84·14	190
200	381·73	1198·4	354·6	843·8	0·4359	2·294	1·0545	84·31	200
225	391·79	1201·4	365·1	836·3	0·4876	2·051	1·0576	84·76	225
250	400·99	1204·2	374·7	829·5	0·5393	1·854	1·0605	85·06	250
275	409·50	1206·8	383·6	823·2	0·5913	1·691	1·0632	85·25	275
300	417·42	1209·3	391·9	817·4	0·644	1·553	1·0657	85·35	300
325	424·82	1211·5	399·6	811·9	0·696	1·437	1·0680	85·49	325
350	431 90	1213·7	406·9	806·8	0·748	1·337	1·0703	85·59	350
375	438·40	1215·7	414·2	801·5	0·800	1·250	1·0724	85·65	375
400	445·15	1217·7	421·4	796·3	0·853	1·172	1·0745	85·61	400
500	466·57	1224·2	444·3	779·9	1·065	0·939	1·0812	85·43	500

* $H_E = P V_E / J$, where P = pressure of steam per square foot, J = Joule's equivalent (778 for 1 B.T.U.), and V_E = the extended volume of steam at pressure, p. Or, $V_E = (V_S - V_w)$ = (the vol. of 1 lb. of steam – the vol. of 1 lb. of water) at the pressures, p, in this Table, as explained in Lecture XI., where the vol. of 1 lb. of water = ·016 cubic foot.

† The figures for this 14·7 lbs. line are the nearest round numbers for easy calculation.

Note.—The specific heat of dry saturated steam at constant pressure, or the heat in B.T.U. required to raise 1 lb. of steam through 1° Fah., is equal to ·48. It is usually taken as being constant at all pressures and temperatures.

Explanation of Sensible and Latent Heats by the Kinetic Theory of Heat.—According to the kinetic theory, heat is a rapid vibratory motion of the ultimate particles of matter, and temperature is the outward manifestation of this motion. An increase or a decrease in the temperature of a body means an increase or a decrease of molecular kinetic energy. Hence, by "sensible" heat is meant that heat which is effective in changing the molecular kinetic energy of the body. The sensible heat given to 1 lb. of water between the temperatures 32° F. and 212° F., is 180 B. T. U., and the whole of this heat is employed in giving a more rapid vibratory motion to the molecules of the water.* The amount of work done in increasing the kinetic energy of the molecules of the water during this change of temperature may be mentally pictured in this way. The sensible heat is equivalent to $180 \times 772 = 138{,}960$ ft.-lbs. of mechanical work, and corresponds to the work done in raising a weight of rather more than 62 tons through a vertical height of 1 foot; or, it is equivalent to the work done in projecting a 5 lb. shot from a gun with a velocity of 1336 ft. per second! The whole of this work, be it remembered, has been done within the mass of 1 lb. of water between the freezing and boiling points. If, then, by any contrivance we convert the whole of the heat given out during the cooling of 1 lb. of water from its boiling to its freezing point, we should be able to do mechanical work to the extent of 138,960 ft.-lbs.

We have shown, that during the conversion of a solid into a liquid, or a liquid into a gas, an amount of heat disappears without in any way affecting a thermometer placed in the mixture; until, the change of state of the whole mass has been completed. Thus, in converting 1 lb. of ice at 32° F. into water, 143 B. T. U. disappear before a change of temperature takes place. In the same way, 966·6 B. T. U. disappear during the conversion of 1 lb. of water at 212° F. into steam at the same temperature. The question may then be asked, what becomes of this heat? Evidently no part of it is employed in increasing the kinetic energy of the molecules of the body, otherwise this would be indicated by an increase of temperature. The older physicists, believing that heat was a substance—a highly elastic, imponderable and subtle fluid, called *caloric*—accounted for the above phenomenon by saying that this "caloric" became *latent* or hidden in some out-of-the-way holes or pores of the body. But we now know that heat is not a substance, and we cannot conceive of any such

* We shall see in Lecture XI. that rather less than 180 B. T. U. are employed in increasing of molecular kinetic energy; but the difference is so small that we may safely neglect it.

cavities or pores in matter wherein this "caloric" could possibly conceal itself in the manner suggested. Further, this disappearance of heat never occurs except when there is a change of state of the body. To clearly understand what actually takes place we require to give a brief explanation of the fundamental differences of the three states of matter as presented to our senses. According to the theory of the *molecular constitution of matter*, the distinctive character of a solid is the fixedness of the molecules relatively to each other. The molecules have a rapid tremulous motion about their mean positions, but are otherwise so firmly bound to their neighbours that work has to be done against the molecular attractions before they can be given greater freedom of movement or separated from each other. Hence, considerable effort is required to separate one portion of a solid from the remainder of the mass. Whenever the molecular attractions are sufficiently overcome, that the molecules glide freely over each other and move about throughout the whole mass, we have all the characteristics of a liquid. The greater the mobility of the molecules the more perfect is the liquid. Hence, the difference between a solid and a liquid is the ease with which the parts of the latter can be separated from each other, and the readiness with which the whole assumes the form of the containing vessel. With gases, on the other hand, the mobility of the molecules is very much greater than in the case of liquids. Here the molecular forces are repulsive, and these cause the molecules to separate from each other as far as the sides of the containing vessel will permit. Thus, a portion of gas, however small, when allowed to enter a vessel, however large, soon diffuses itself equally throughout the whole vessel, and this is true whether there are other gases or not in the vessel along with it.

We are now in a position to understand what becomes of the so-called *latent* heat. In converting a solid into a liquid, or a liquid into a gas, work has to be done in effecting certain molecular actions, as in overcoming the molecular attractions characteristic of solid substances, or bringing into play those molecular repulsions characteristic of the gaseous state. Hence, during those transient states of matter, the so-called latent heat disappears as heat, but reappears as the result of molecular mechanical work.

As before, we may give a mental picture of the vast amount of work done within the mass of 1 lb. of water during those physical changes.

In converting 1 lb. of ice at 32° F. into water at the same temperature, 143 B. T. U. (or, $143 \times 772 = 110{,}396$ ft. lbs. of work) have been expended against the molecular attractions.

This corresponds to the work done in raising a weight of about $49\frac{1}{2}$ tons through a vertical height of 1 foot; or the work done in projecting a 4 lb. shot from a gun with a velocity of about 1330 feet per second!

In converting 1 lb. of water at 212° F. into steam at the same temperature, 966·6 B. T. U. (or, $966{\cdot}6 \times 772 = 746{,}215$ ft. lbs. of work) are expended in bringing about this physical change. This corresponds to the work done in raising a weight of rather more than 333 tons through a vertical height of 1 foot; or the work done in projecting an 18 lb. shot from a gun with a velocity of more than 1600 feet per second!

Conversely, when 1 lb. of steam at atmospheric pressure (212° F.) is condensed into water at the same temperature, the work done by the colliding or "clashing" of the molecules corresponds to 746,215 ft. lbs. or 966·6 B. T. U.*

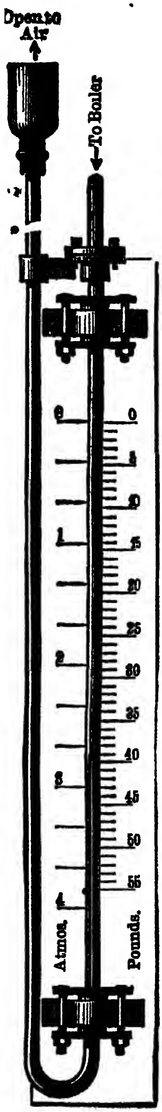
If, then, an engine could convert the whole of the heat given out during the reduction of steam at 212° F. into water at the same temperature, mechanical work to the extent of 746,215 ft. lbs. would be done per. lb. of steam condensed. We shall afterwards see that only a very small fraction of this work can be made use of in practice.

* We shall see in Lecture XI. that the whole of the 966·6 B. T. U. are not employed in molecular work; for about 73 B. T. U. go to perform work *external* to the substance.

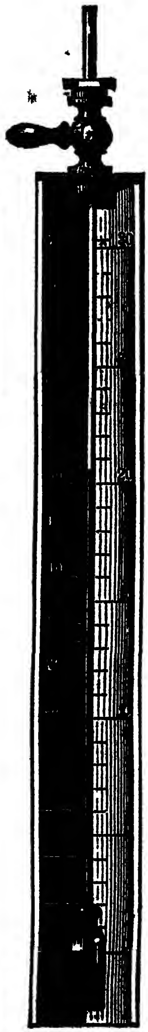
Pressure Gauges.—Instruments for indicating the intensity of the pressure of a fluid contained in a closed vessel, are called "pressure gauges," or "vacuum gauges," according as they register how much the pressure is above or below that of the atmosphere.

The Mercurial Pressure Gauge, as seen by the first figure, consists of a bent, U, glass tube, containing mercury, from, O, round the bend of the tube to, O. One end is connected directly to the closed vessel, or say to a steam boiler, while the other end is connected to a cup, to prevent the mercury being lost when the pressure rises higher than the range of the tube. This cup is open to the air, and consequently the pressure of the atmosphere acts on that side of the mercurial column. A vertical scale is fixed immediately behind the vertical limb connected to the boiler or closed vessel, and it is graduated in any convenient manner—say, for lbs. per square inch of pressure. As the pressure increases, the mercury in this limb is depressed, and rises correspondingly in the other limb. When the pressure in the closed vessel equals that of the atmosphere, both ends of the mercury should stand at 0. The pressure as shown by the right hand scale is 39 lbs., and by the left one as fully 2¾ atmospheres. Nothing could be simpler or more accurate than this arrangement, for, as we saw in the case of the Marcet's boiler, a vertical column of mercury produces a definite pressure of about 1 lb. per square inch for every 2 inches in height. In practice, however, the inside of the glass tube gets coated with a dirty film, owing to the oxidation of the mercury, which prevents the attendant observing the exact position of the depressed end of the mercurial column.

Such a pressure gauge is, of course, inadmissible on board a ship or on a locomotive, owing to the jerking motion;



PRESSURE GAUGE.


VACUUM GAUGE.

and further, the length of the tube would have to be very great for the pressures now carried in high-pressure steam boilers (about 300 inches, or 25 feet for 150 lbs. on the square inch). For these reasons its use has been discarded in ordinary practice; but, as an exact and standard instrument for scientific purposes, and for testing and calibrating the working pressure gauges (which we are about to describe), it is indispensable. In all the best works where ordinary pressure gauges are made and tested, a long graduated vertical mercury column or gauge is supplied, with which these may be compared; and there, the inside of the glass is occasionally rubbed clean by a little cotton-wool fastened to the end of a wire and dipped in sulphuric acid.

Mercurial Vacuum Gauge.—This gauge indicates directly the *absolute* pressure inside a vessel such as the condenser of a steam engine, the suction pipe to an air-pump, or the vacuum pan of a sugar-refinery. The simplest form is shown by the second figure on the previous page. It consists of a vertical glass tube a little over 30 inches in length, with its lower end open and dipping into mercury contained in an iron bottle, while its upper end is attached to a brass cock and pipe connected with the vessel or condenser. A scale is fixed behind the glass tube graduated on the right hand into inches, and on the left hand into millimetres, but it would be more convenient if this latter scale were divided so as to show the absolute or the back pressure in lbs. per square inch due to an imperfect vacuum. The more perfect the vacuum, the higher the mercury rises in the tube, due to the atmosphere pressing on the mercury through a small hole near the top of the iron bottle. Every 2 inches of rise corresponds to a diminution of about 1 lb. of back pressure per square inch.

It does seem absurd that we should thus continue to register pressures in three or four different ways.

1. In lbs. per square inch above the atmosphere—*e.g*, in the case of the pressure of steam in a boiler by ordinary steam gauges.

2. In inches of mercury from atmospheric pressure downwards, towards a perfect vacuum, or in lbs. per square inch below atmospheric pressure—*e.g.*, in the case of ordinary vacuum gauges attached to condensers.

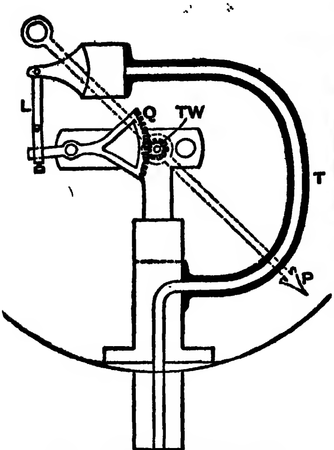
3. In lbs. per square inch reckoned from a perfect vacuum, or what are termed lbs. per square inch absolute—*e.g.*, in the case of the back pressure during exhaust of a condensing engine.

If we universally adopted the last of these methods, there would be no confusion, and only one way of reckoning pressures--viz., from absolute zero. Condenser vacuum pressures would then range from 0 to 15 lbs., and boiler pressures from 15 lbs. upwards.

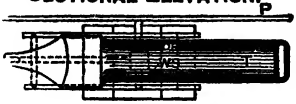
Bourdon's Pressure and Vacuum Gauges.—Steam pressures in boilers or pipes are usually indicated by Bourdon's pressure gauges, and negative or vacuum pressures in condensers, &c., by Bourdon's vacuum gauges, or by instruments of somewhat similar design and construction.

The construction of Bourdon's pressure gauges is clearly shown by the figures on the opposite page. Figure C shows the internal mechanism of such a gauge in its earliest and simplest form. Figures A and B show a sectional elevation and plan and a front outside view of a modern high-pressure gauge as made by Messrs. Schäffer & Budenberg. The internal mechanism of this modern form of gauge differs from the older forms only in the arrangement of details, so as to give correct readings at high pressures. The action of the gauge is as follows:—The steam, water, or gas enters by the cock (shown with the gauge in figure B) to the curved metallic tube, T. This tube is made of hard brass or steel, and has its upper end hermetically sealed. The cross-section of this tube is of a flat oval form, as

A

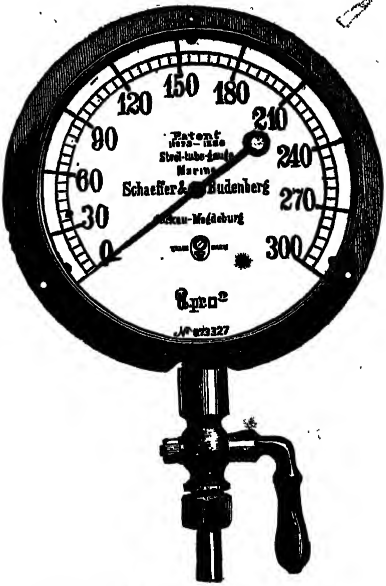


SECTIONAL ELEVATION.



PLAN.

B



FRONT VIEW OF HIGH-PRESSURE GAUGE.

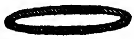
C

BACK VIEW OF EARLIEST FORM OF BOURDON GAUGE.

INDEX OF PARTS.

T represents Tube.
L ,, Link.
Q ,, Quadrant.
T W ,, Toothed Wheel.
D P ,, Distance Piece.
P ,, Pointer.

D



Section of Tube.

shown in figure D. It has its greatest breadth fixed perpendicularly to the direction in which the tube is curved. When the pointer is at zero (that is, when the pressure inside the tube is the same as that outside) the tube is in its normal state as regards shape and curvature. If, now, the pressure of the fluid contained in the tube be above the external or atmospheric pressure, the tube tends to become more and more circular in cross-section the greater the pressure within it, and the tube as a whole tends to straighten out, thereby pulling the link, L, upwards. This motion is transmitted to the quadrant, Q, and thence to the toothed wheel, T W, fixed on the same axis as the pointer, P, which latter moves over the scale.

Should the pressure within the tube be less than that of the surrounding atmosphere (as is the case when the instrument is measuring the vacuum of a condenser), then the cross-section of the tube becomes flatter than its normal or ordinary shape, and, consequently, the closed end of the tube curves inwards, thus moving the pointer, P, in the other direction.

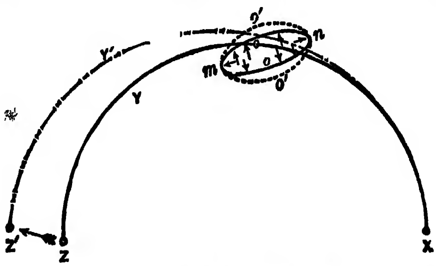


DIAGRAM SHOWING THE ACTION OF THE BOURDON TUBE UNDER PRESSURE.

The discovery of the peculiar action of the tube in a Bourdon gauge is exceedingly interesting. M. Bourdon was engaged trying to restore to its original circular section the worm-pipe of a still which had become flattened. He endeavoured to do so by pumping water into the pipe at a great pressure, knowing that the section of the pipe would conform to the shape consistent with greatest strength, which, of course, is the circular shape. During the experiment, however, it was observed that not only did the shape of the cross-section tend to change, but the tube as a whole tended to straighten out. The observation of these facts thus led M. Bourdon to usefully apply a bent oval tube in the form of a gauge for measuring fluid pressure.

The following is an explanation of the action which takes place:—Let X Y Z represent the curve of the tube in its normal state, Z, being the free or movable end. Let *m o n* represent the cross-section of the tube when the pressure inside is balanced by, or equal to, that outside. It is shown by higher geometry that when a flexible and inextensible surface is bent simultaneously in two directions at right angles to each other, that the product of the *curvatures*,* in these directions, *is a constant quantity*. Now

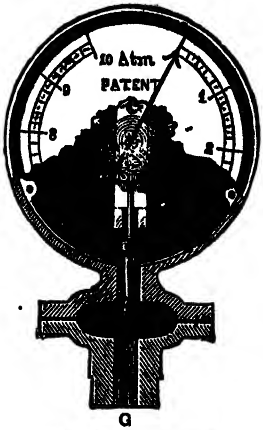
* The *curvature* of a line at any point is the reciprocal of the *radius of the curve* at that point. The *curvature* of a circle is constant and equal to $\frac{1}{r}$, where r is the radius of the circle.

the tube of a Bourdon's gauge is bent in such a manner—for it is bent in the direction of its length, and also at right angles to this direction. Let r_1, r_2 be the radii of these two curves, the latter being the curve of the cross-section. Then by the above theorem,

$$\frac{1}{r_1} \times \frac{1}{r_2} = \text{constant.}$$

From this equation it follows that when one factor of the term on the left-hand side increases the other factor must decrease, and conversely. Now this is really what happens. When the pressure of the fluid inside the tube is increased the cross-section tends to become rounder, like the dotted lines, *m o′ n*, that is, the *radius of the curve*, r_2, becomes *less*. Consequently, the *curvature* is greater than before, so that the curvature, $\frac{1}{r_1}$, of the tube in the direction of its length is less than before, which means that the tube has straightened out into some position such as X Y′ Z′. The pointer is thus moved to positions which have been marked by trial to correspond with the excess of pressure inside the tube above that outside. The reverse of this takes place when the pressure within the tube is less than that outside—*i.e.*, the curvature of the cross-section becomes less, whilst that of the tube becomes correspondingly greater.

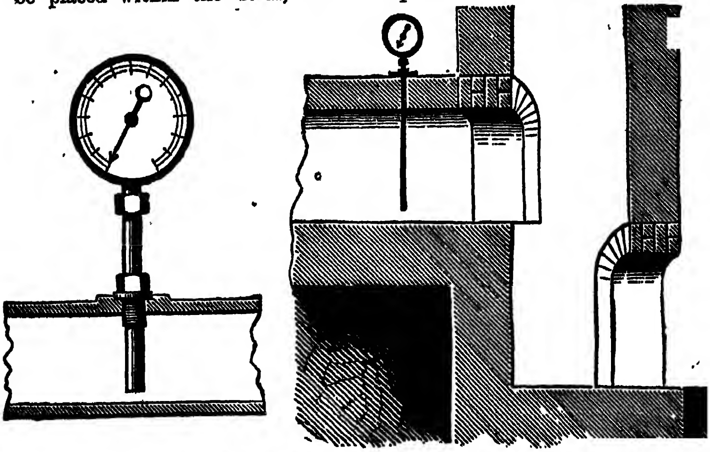
Schäffer's Gauge. — Another form of gauge, which represents Schäffer's patent, is shown in the accompanying figure, the difference between it and the Bourdon patent being, that the former relies upon the natural elasticity of a concentrically corrugated steel plate placed across the hollow opening in the flange of the pipe, G, which communicates with the boiler or condenser. The centre of this corrugated plate is attached by a clip and rod to the toothed quadrant as shown. When the pressure is greater than that of the atmosphere, the corrugated plate is bulged upwards, and when it is less, it is bulged downwards. These motions, being proportional to the pressures per square inch, are correspondingly indicated on the graduated dial by the pointer.



SCHÄFFER'S GAUGE.

Pressure Pyrometer.—In Lecture III., under the heading of Pyrometers, we referred to this instrument, which depends for its action upon the fact that the pressure of a gas, generated from a liquid with which it is in direct communication, corresponds to the temperature of the liquid. The name given to it by the makers is the **Thalpotasimeter**, and it is constructed, as may be seen from the following figures, of a metal stem, containing the liquid, and ending in a Bourdon or Schäffer gauge. The metal stem is shown fixed in the first case into a pipe, and in the second case into a flue, through which hot gases are passed. Their temperature inside the pipe or the flue is communicated to the

stem of the instrument, and therefore to the liquid within it. If wat be placed within the stem, then the pressure (and consequently tl



PRESSURE PYROMETERS OR THALPOTASIMETERS.

temperature) rises in accordance with Regnault's tables. Instruments filled with ether are made and graduated from 100° to 220° Fah.; those filled with water, from 212° to 680° Fah.; and those filled with mercury, up to 1,400° Fah.

Lecture VII.—Questions.

1. What is the distinction between sensible and latent heat?

2. Describe an experiment by which you could show that heat becomes latent when water is converted into steam.

3. What is meant by saying that the latent heat of steam is 966·6? Point out the sources of error in Black's experiment when he tried to find the latent heat of steam.

4. How would you ascertain the pressure of the vapour of water at a temperature above 212° F.? Describe some method of conducting the experiment.

5. When you speak of the "latent heat of steam," what property of steam do you refer to? State the numerical value of the latent heat of steam at 212° F. A pound of water at 212° F. is passed into 20 lbs. of water at 70° F., what is the final temperature? *Ans.* 76°·7.

6. From the table of Regnault's results, plot out a curve showing the rise in pressure of steam with increase of temperature.

7. How many pounds of ice at 32° F. will be converted into water at 40° F. by mixing it with 6 lbs. of water at 160° F.?

6 lbs. of water *gives* up 6 (160° − 40°) = 720 units.

1 lb. of ice *takes* up 143 + (40 − 32) = 151 ,,

∴ 720 ÷ 151 = 4·768 lbs.

8. Into 1 cwt. of water at 45° F. is poured 20 lbs. of water at 160° F., and then 4 lbs. of ice at 32° F. are added. What is the final temperature when the ice has just melted?

Water 112 (45° − 32°)	=	+ 1456	units of heat from 32° F.
Water 20 (160° − 32°)	=	+ 2560	,, ,,
Ice 4 × 143	=	− 572	to convert 4 lbs. ice into water.
Total 136 lbs. mixture	=	3444	units left.

∴ 3444 ÷ 136 = 25·32 above 32° or 57°·32 F.

9. If the heat which just melts 8 lbs. of ice at 32° F. were applied to 30 lbs. water at 60° F., to what temperature would the water rise?

8 × 143 = 1144 units of heat required to melt the ice.

Now, 30 lbs. of water raised 1° F. = 30 units of heat,

∴ 1144 ÷ 30 = 38°·13 F. of rise from 60° F. or 98°·13 F.

10. There are mixed together 200 lbs. of water at 100° F., 3 lbs. steam at atmospheric pressure, and 15 lbs. of ice at 32° F. What is the resulting temperature when all the ice is just melted?

The 200 lbs.	water	has	+	13,600 *u*,	more than	water at	32° F.
,, 3 ,,	steam	,,	+	3,438	,,	,,	,,
,, 15 ,,	ice	,,	−	2,145	less	,,	,,
∴ 218 ,,	mixture	,,		14,893	more	,,	,,

And 14,893 ÷ 218 = 68°·3 F. above 32 = 100°·3 F.

11. Define the terms *latent heat, foot-pound, thermal unit.* Write down the number which expresses the latent heat of steam at 212° F., and explain how that number is arrived at.

12. When does heat become latent? What do you understand by the expression *latent heat of steam?* What unit is adopted for measuring and comparing quantities of heat? Write down the number expressing the latent heat of steam at 212° F.

13. What is the thermal unit employed in this country? State its measure in foot-pounds. How many thermal units are expended in converting one pound of water at 60° F. into one pound of steam at 212° F.? *Ans.* 1,118·6.

14. Distinguish between the sensible and latent heat of steam. How many thermal units must be added to 1 lb. of water at 32° F. to raise it to 212° F. and evaporate it into steam? How many of these units go to sensible and how many to latent heat? *Ans.* 1,146·6 B.T.U.; 180 B.T.U.; 966·6 B.T.U.

15. Write a brief essay explaining what you consider "sensible" and "latent" heat to be; and illustrate the same by means of one or two examples.

16. Prove the well-known law connecting latent heat, temperature, pressure, and change of specific volume when there is change of state. From the following figures find approximately the volume in cubic feet of a pound of steam at 293° F. :—

Temperature, Fahr.	Pressure in Lbs. per Square Inch.	Latent Heat, in Fahr. Heat Units.
284°	52·52	915·1
293°	60·40	908·6
302°	69·21	902·2

(S. & A., 1897, Hons.)

17. Given a table of temperatures and densities of saturated steam, how would you, by means of squared paper, test if there are empirical laws like these :—Volume of 1 lb. $= a p^n$; total heat $= c + e\iota$? What are the values, which are usually taken to be correct, for a, n, c, e? (S. & A., 1898, Adv.)

18. Given the following temperatures and pressures in lbs. per square foot :—284° F., $p = 7{,}563$; 293° F., $p = 8{,}698$; find $\frac{dp}{dt}$ roughly. Take the well-known formula for latent heat, assuming the specific heat of water to be constant, and calculate approximately the volume of 1 lb. of steam at 288½° F. Prove the rule you apply. (S. & A., 1898, H., Part i.)

19. Answer only *one* of the following (*a*) or (*b*) :—Describe how you would experimentally determine—(*a*) The calorific value of any kind of heating gas or oil; (*b*) the latent heat of steam, say at about 120° C. (B. of E., 1901, Adv. and Hons., Part i.)

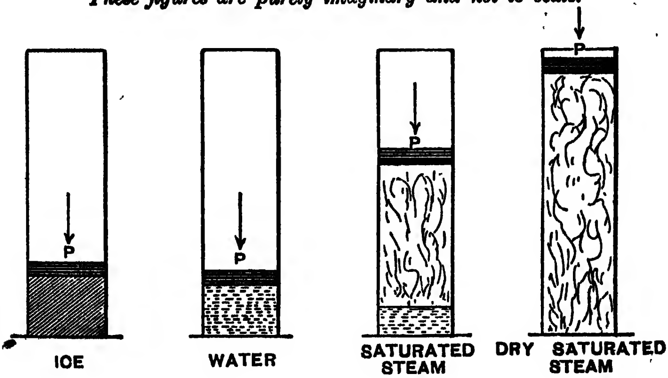
20. Suppose you were requested to find the temperatures in the fire-box and chimney of a locomotive, or in the furnace and uptake of a marine boiler funnel, how would you do it? Give arithmetical examples in each case.

LECTURE VIII.

CONTENTS.—The successive effects produced by the continuous application of Heat to a piece of very cold Ice until Dissociation takes place—The boiling point of a Liquid—Experiment of Water boiling at pressures less than one Atmosphere—Use of large Air Pumps in connection with Condensers—Total Heat of Evaporation.—Questions.

WE shall best understand the physical properties of steam and the results arrived at by Regnault, by considering, in the first place, the several changes which take place in water—from its solid condition, ice, until it becomes dissociated under the continuous application of heat.

These figures are purely imaginary and not to scale.



Referring to the figure, suppose that we put 1 lb. of very cold ice in the bottom of an open-mouthed cylinder, and place a piston on it, which, together with the pressure of the atmosphere, exerts a pressure of p lbs. on the square inch.

STAGE 1.—On the application of heat to the bottom of the cylinder, the ice is gradually heated until it arrives at 32° F.

STAGE 2.—The temperature now remains constant until all the ice melts and becomes converted into water. The bulk of the water being less than that of the ice from which it is formed, the piston descends a very little. As we have already noticed

in Lecture VII., 143 units of heat must be communicated to the 1 lb. of ice at 32° F. before it is all melted into water at 32° F.

STAGE 3.—Still applying heat, the water increases in temperature while the bulk diminishes, until 39° F. is reached (the maximum density point of water); thereafter, the volume gradually increases, but in a very slight degree, with the rise in temperature, until a little above 212° F. is reached, the limiting temperature of the water depending on the pressure, p, lbs. on the square inch (see Regnault's tables). Had the pressure on the piston been nothing more than that due to the normal pressure of the atmosphere, viz., 14·7 lbs., corresponding to a barometric height of 29·9 inches, then the water would have been converted into steam at a temperature of 212° F.

STAGE 4.—The temperature remains stationary at that limit value, and the formation of steam commences, the piston rising as more and more of the water is evaporated. So long as any water remains at the bottom of the cylinder, we are producing what is called *saturated steam*, or wet steam. This is the condition of steam usually supplied to engines from ordinary boilers having small steam space or no steam dome.

STAGE 5.—When all the water in the bottom of the cylinder has been evaporated, and *just* when all the water or aqueous particles held in suspension with the steam have been converted into steam, we obtain *dry steam*, or what is sometimes termed *dry saturated steam;* then 966·6 units of heat must have passed into the contents of the cylinder, for, as we have already noticed in Lecture VII., 966·6 units of heat must be communicated to the 1 lb. of water before it is all converted into steam at 212° F. The ratio of the weight of dry steam to the total weight of steam and water is termed the *dryness fraction*. If x be the weight of dry steam in 1 lb. of wet steam, then $(1-x)$ is the weight of water held in suspension. When $\frac{9}{10}$ of the vapour is steam, then $\frac{1}{10}$ is water, or 10% of water is in suspension.

STAGE 6.—If more heat be added to the dry steam in the cylinder, the pressure, P, on the piston remaining the same, the temperature will again begin to increase, and we get what is termed *superheated steam*. The more it is heated, the more nearly do its properties approach to those of a perfect gas. If the top of the cylinder had been closed from the commencement of stage 3, the pressure would have risen with the temperature until the commencement of stage 6, in accordance with Regnault's tables, given at the end of last Lecture; but during stage 6 we communicate more heat to the steam than its pressure would indicate by the tables. Superheated steam up to 500° or even 600° F. is now used for engines. The various reasons for its

rapid adoption are given later on in Lecture XIV., &c. (see Index).

Stage 7.—Steam cannot be heated indefinitely without a molecular change taking place, termed *dissociation*, when it separates into constituent gases—hydrogen and oxygen. This action is practically carried out in the process of making "water gas," by blowing dry steam over very hot plates before carbonising it, ready for illuminating purposes.

Thus the successive effects produced by the continuous application of heat to a piece of very cold ice are:—

1. Heating ice up to 32° F.
2. Melting ice, absorption of latent heat, 143 units per lb.
3. Heating water up to boiling point.
4. Formation of saturated steam, no increase of temperature.
5. Formation of dry steam, due to the complete absorption of the latent heat, or 966·6 units per lb. of water.
6. Superheated steam, increase of temperature above stage 3.
7. Dissociation or formation of hydrogen and oxygen.
8. Heating, no further alteration of the physical state.

Boiling Point.—Before treating of the "total heat of evaporation," we shall digress a little to consider what is meant by the boiling point, or the temperature of ebullition.

The boiling point of any liquid is that point on the temperature scale, when the tension throughout its mass just overcomes the surrounding pressure. The temperature of the boiling point, therefore, depends directly on the *pressure* under which the liquid is evaporated, and the greater the pressure the higher the temperature at which it boils.

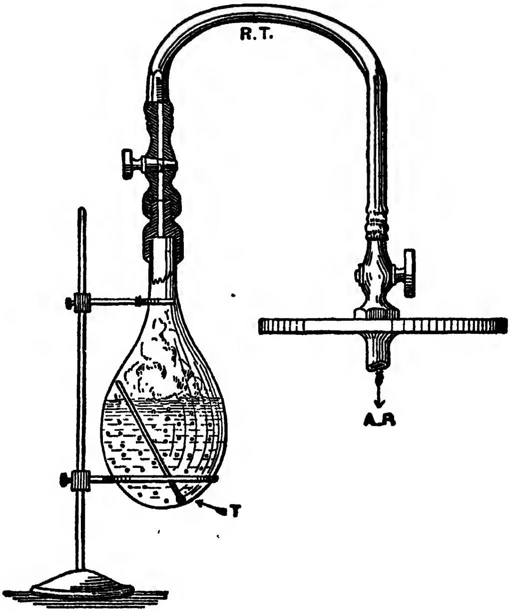
The boiling points of fresh water at different pressures are approximately as follows (compare this with Regnault's table in Lecture VII.):—

Under a pressure of $\frac{1}{8}$ atmosphere,	123° Fah.
	150°
	179°
	212°
	249°
	273°
	291°
	306°
	319°

It is thus clear that water will boil or give off steam far below, as well as far above, its normal boiling point, 212° F.

To illustrate this take a glass flask half full of water with a thermometer in it, heat it over a spirit lamp or Bunsen burner,

until the water just begins to boil and the temperature, as registered by the thermometer, is 212° F. Now attach it, as shown, to an air-pump, A P, by a flexible india-rubber tube, and begin extracting the air. The water is observed to boil violently, although it may have cooled down to as low as 180° F. This plan of attaching it to the air-pump is much better than that of placing it under the glass-bell jar of the pump, as it permits the thermometer being easily seen after the moist steam has begun to rise.



If an air-pump is not at hand, the following simple experiment will illustrate the fact equally well to a class:—After heating the water in the flask to 212° F., and letting it boil freely for a minute to expel the air, cork it up quickly and tightly, leaving a thermometer inside. Now pour cold water on the outside of the flask, the water will at once begin to boil, although the temperature may be now below 200° F. It ceases to boil, however, if you stop cooling it. Why? Because the tension of the vapour generated equals that of the natural tension of the water; but condense this vapour by a second application of cold

water, and again it begins to boil, even with the temperature below 180° F. A knowledge of these facts is most important to the engineer, for it shows him that in the condensers of large engines, he must provide air-pumps of sufficient capacity to carry off the steam vapour generated at even low temperatures. It was only last year that an acquaintance of the author's, over-

looking this point, put in a set of very small air-pumps to a pair of marine engines which he was constructing, under the impression (due to miscalculation) that all that was necessary was to lift the condensed water, and that marine engineers generally, were putting on air-pumps out of all proportion to the work to be done! He soon discovered his mistake, for, on the day of the trial trip, he could not keep up a vacuum above a few inches. In addition to the steam vapour which is generated at pressures below the atmospheric pressure, any air which may have come over with the steam at once expands on a reduction of pressure, and has to be sucked away at every stroke, otherwise it will spoil the vacuum. We shall refer to this point again when we come to treat of condensers and air-pumps.

The experiment of raising the boiling point by raising the pressure is easily done. Procure a flask, as in the former experiment, with a tight-fitting stop cock. Half fill the flask with water, heat it with the cock open until the water boils and all the air has been expelled, then shut the stop cock. The steam now generated rises in pressure and temperature. The increasing pressure raises the boiling point and thus stops the violent ebullition, unless heat is applied very rapidly. Allow the temperature to rise, say to 240° F., then slightly open the cock, ebullition is at once observed, although the pressure is equal to two atmospheres above a perfect vacuum.

Lecture VIII.—Questions.

1. Describe in your own words the several effects which take place in succession on applying heat to a lump of ice enclosed in a cylinder. Distinguish between saturated steam, dry saturated steam, and superheated steam.

2. How much ice at 0° C. will be converted into water at 5° C. by mixing it with 10 lbs. of water at 80° C.? *Ans.* about 9 lbs.

3. The latent heats of 1 lb. of water and 1 lb. of steam are respectively 143 and 966·6 according to the Fah. scale; work out in full by proportion what they are according to the Cent. scale. *Ans.* 79·4 and 537.

4. How many British units of heat are required to raise 1 cubic foot of water (62·5 lbs.) from 15° C. to 100° C.? *Ans.* 9562·5.

5. What is the resulting temperature on mixing 20 cubic feet of water at 212° F. with 100 cubic feet at 10° C.? *Ans.* 77° F.

6. The diameter of a cylinder is 20 inches, steam is admitted at a pressure of 100 lbs. on the square inch; what is the total pressure in lbs.? *Ans.* 31,416 lbs.

7. Steam blows off at 60 lbs. pressure from a boiler by gauge, the barometer stands at 29 inches; what is the temperature of the water in the boiler? *Ans.* 307°·4 F.

8. Account for the use of larger air-pumps being used with condensing engines than would merely suffice to lift the weight of water in the condenser.

9. What is meant by the "dryness fraction," and how is it estimated?

10. How many units of heat would be absorbed in raising 18 lbs. of steam of atmospheric pressure from water at 65° F.? *Ans.* 20,044.

11. How much water at 55° F. could just be brought to the boiling point by the latent heat given up by 2 lbs. of steam at atmospheric pressure being condensed? *Ans.* $(966 \times 2) \div 157 = 12{\cdot}3$ lbs.

12. What are *saturated*, *superheated*, and *wet* steam respectively? Why is there a loss of efficiency in using wet steam? Define a thermal unit, and explain the method of measuring the latent heat of steam.

13. If one pound of Newcastle coal develops 12,000 units of heat by its complete combustion, how much water at 60° F. should be converted into steam at 212° F. by the consumption of 1 cwt. of such fuel, assuming that there is no loss of heat during the operation? *Ans.* 1,201·5 lbs.

14. At an Electric Power Station, 4,150 units of electric energy were sold in 24 hours, the coal consumed being 16,200 lbs. And, on another occasion, 2,489 units were sold in the 24 hours, the coal consumption being 12,380 lbs. It is known that, if units of electricity and weight of coal are plotted on squared paper, the points will lie fairly well on a straight line. The maximum output is 25,000 units. Find the coal consumed in the 24 hours when there are the daily outputs of 8, 16, 24, and 50 per cent. of the maximum. In each case what is the coal per unit? Tabulate your answers. (B. of E., 1902, Adv.)

LECTURE IX.

CONTENTS.—Total Heat of Evaporation—Quantity of Water required for Condensation of Steam, with Examples—Questions.

Total Heat of Evaporation.—The total heat of evaporation is the sum of the sensible and the latent heats of evaporation, and is approximately a constant quantity for pressures near the atmospheric pressure.

The heat required to elevate the temperature of 1 lb. of water from the freezing point, 32° F., to the temperature of evaporation, is called the *sensible heat*,* and the additional heat required to evaporate it is termed the *latent heat* (see Lecture VII.)

The total heat of evaporation for water is, therefore, the quantity of heat in thermal units necessary to raise 1 lb. of water from the freezing point, 32° F., to the particular temperature at which it is being evaporated, and to evaporate it at that temperature.

Let H stand for the Total heat of evaporation in B.T.U.
S " " Sensible heat " ".
L " " Latent heat " "

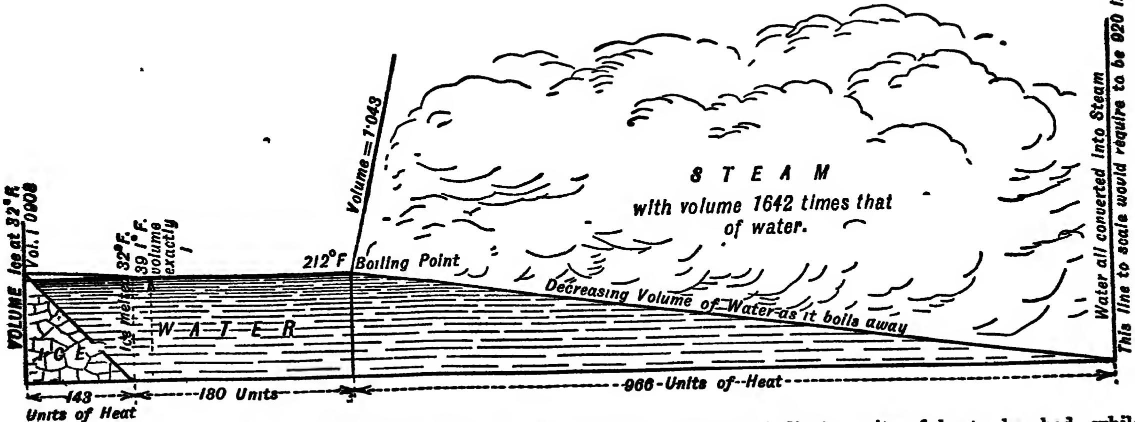
$$\text{Then, } H = S + L.$$

Now, since we have defined a unit of heat to be the quantity of heat necessary to raise 1 lb. of water by 1° Fah., the amount of heat imparted to 1 lb. of water, in raising its temperature from 32° F. to 212° F., must be $(212 - 32) = 180$ such units.† Therefore the *sensible heat* of steam at 212° F., is said to be 180 units per lb. or 180. Again, we saw, Lecture VII., that the *latent* heat of steam at 212° F. was in round numbers 966 units per lb., or 966.

* The reason for starting from the freezing point of water, and not from zero Fah., is that we thus avoid the introduction of the latent heat of water. Of course, if the water is of higher temperature than 32° F. to start with, the heat required to be applied to it to bring it up to the point of evaporation is correspondingly less.

† We here neglect, for the sake of simplicity, the addition to our former definition of this unit (see Lecture IV.)—"water at its maximum density point," and, therefore, the very slight difference in the sensible heat caused by the increase of the specific heat of water as it rises in temperature. This difference simply amounts to that between 180 units and 180·9.

GRAPHICAL REPRESENTATION OF THE CHANGES FROM ICE INTO WATER, AND WATER INTO STEAM AT ATMOSPHERIC PRESSURE, DUE TO THE ABSORPTION OF HEAT, WITH THE CORRESPONDING TEMPERATURES AND VOLUMES.



Explanation of Diagram.—Distances measured horizontally from the left indicate units of heat absorbed, while distances measured vertically indicate volumes. We commence with ice occupying a volume of 1·0908. The application of heat to the ice (which is supposed to be at 32° F.) immediately begins to melt it, and when 143 units per lb. have been absorbed, the whole of the ice is melted, and we have water occupying a volume 1·000127. The further application of heat causes the volume of this water first to decrease to 1 (at a temperature of 39°·1) and then again to increase to 1·043 at boiling point. After this, the application of each unit of heat causes $\frac{1}{966}$ part of the water to pass away as steam, and when 966 units per lb. have been absorbed the whole of the water has passed into steam, which now occupies a volume 1642 times that of the water from which it was produced.

Therefore the *total heat* of steam at that temperature must be—

$$\begin{aligned} H &= S + L \\ &= 180 + 966 \\ &= 1{,}146 \text{ Units of Heat.} \end{aligned}$$

If steam is generated at a higher temperature than 212° F., the sensible heat increases, and the latent heat decreases.

The following formula, deduced from Regnault's experiments, gives approximately the *latent heat* of steam produced at other temperatures Fah. :—

$$L = 966 - 0{\cdot}7\,(t^\circ - 212^\circ).$$

Where L, is the latent heat, and t°, the temperature of evaporation on Fahrenheit scale.

Consequently, the total heat of evaporation, at any temperature, t°, must be—

$$\begin{aligned} H &= S + L \\ &= (t^\circ - 32^\circ) + 966 - 0{\cdot}7\,(t^\circ - 212^\circ) \\ &= 1{,}082{\cdot}4 + 0{\cdot}3\,t^\circ. \end{aligned}$$

For example.—Let us find from this equation the total heat of steam at 212° F. Then $t^\circ = 212^\circ$ and $\cdot 3 \times 212 = 63{\cdot}6$, which, added to $1{,}082{\cdot}4 = 1{,}146$, the same as before and in Table II.

The relation between the above heat quantities may be still further studied by comparing the following *Heat Family Tree* with Table II. in Lecture VII. and Example I. in Lecture XI.:—

Total Heat Expended,
H = S + L.

Increase of Sensible Heat, S — Latent Heat of Evaporation, L

Latent Heat of Evaporation, L:

Internal Work Done, or increase of internal energy, during evaporation, $H_I = L - H_E$. — External Work Done during evaporation, $H_E = \dfrac{P\,V_E}{J}$, or, $H_E = \dfrac{144\,p\,(V_S - V_w)}{778}$.

S + H_I:

Internal Energy, $E_I = S + H_I$.

For example, select from Table II. any pressure to test the above equations. Let p = 100 lbs. absolute, then we see from pp. 125, 126, that—

p	H	S	L	V_S	H_E
100	1,181·9	297·9	884	4·403	81·17

Here, $H = S + L = 297{\cdot}9 + 884 = 1{,}181{\cdot}9$ B.T.U.

And $H_E = \dfrac{144\,p\,(V_S - V_w)}{778} = \dfrac{144 \times 100\,(4{\cdot}403 - {\cdot}01605)}{778} = 81{\cdot}17$ B.T.U.

$\therefore\ H_I = L - H_E = 884 - 81{\cdot}17 = 802{\cdot}83$ B.T.U.

And $E_I = S + H_I = 297{\cdot}9 + 802{\cdot}83 = 1{,}100{\cdot}73$ B.T.U.

But $H = H_E + E_I = 81{\cdot}17 + 1{,}100{\cdot}73 = 1{,}181{\cdot}9$ B.T.U., *as above.*

Lecture IX.—Questions.

1. If a pound of water at 212° F. be mixed with x pounds of water at 60°, what is the value of x when the resulting temperature is 120°? Again, if a pound of steam at 212° F. be mixed with y pounds of water at 60°, find y when the resulting temperature is 120°. Account for the difference between x and y. *Ans.* $x = 1·53$ lbs.; $y = 17·6$ lbs.

2. What is the latent heat of steam? If a quantity of steam weighing one pound, and at temperature of 212° F., is condensed in 100 lbs. of water at 60° F., what is the resulting temperature? *Ans.* 71°·06.

3. In a jet condenser the temperature of the condensing water is 60° F. and that of the entering steam is 193° F. Also the condenser remains at a temperature of 120°. Under these conditions find the weight of condensing water per pound of steam which enters the condenser. *Ans.* 17·53 lbs.

4. If there pass at the same time into the condenser, and from thence into the hot-well, 2 tons of water at 55° F. and 1·5 cwt. of steam at atmospheric pressure, what will be the resulting temperature? *Ans.* 95°·6 F.

5. Hot-well 105° F., injection 53°, and steam at atmospheric pressure. Required number of pounds of steam condensed by 4 cb. ft. of the injection water. *Ans.* 12·1 lbs.

6. From 1886 Steam Examination. Temperature of injection water 60° F., temperature of hot-well 100° F., latent heat of exhaust steam 1016 units, its temperature being 140° F.; find the pounds of injection water required per pound of steam condensed. *Ans.* 26·4 lbs.

7. The cylinder of an engine is 74 inches in diameter, and the stroke is 5·5 feet; what is the capacity of the cylinder in cubic feet? How many pounds of water must be evaporated in order to fill the cylinder with steam at a pressure of 15 lbs. absolute (atmospheric pressure), it being given that steam of that pressure occupies 1,642 times the volume of the water from which it is generated? *Ans.* 164·27 cubic feet; 6·24 lbs.

8. An engine working without expansion has a piston of 144 square inches in area with a 12-inch stroke, and the number of double strokes per minute is 60. Steam is supplied at a temperature of 293° F., (the volume of 1 lb. of steam at 293° F. being 7 cb. ft.), find units of heat required per minute for steam from water at 60° F. *Ans.* 19,753·7.

9. Define the term "*latent heat,*" and distinguish between the heat expended in external work and that expended in internal work during evaporation. Find in foot-pounds the external work done in converting 1 lb. of water at 212° F. into steam at 212° F., having given volume of 1 lb. of water = ·016 cubic foot. Volume of 1 lb. of steam at 212° F. = 26·4 cubic feet. Pressure of steam at 212° F. = 14·7 lbs. per square inch. Find also the internal work done. *Ans.* 55,849·65 ft.-lbs.; 690,365·55 ft.-lbs.

10. A unit of heat is sometimes expressed in thermal units and sometimes in foot-pounds. Explain the meaning of this distinction. How many foot-pounds of work are done in converting one pound of water from a temperature of 104° F. into steam at 328° F. corresponding to an absolute pressure of 100 lbs. per square inch? If the volume of 1 lb. of steam, at the above temperature and pressure, be 4·33 cubic feet; find external work done during formation, and weight of steam used per hour per horse-power. *Ans.* 1,108·8 B.T.U. = 855,993·6 ft.-lbs.; 62,352 ft.-lbs.; 31·04 lbs.

LECTURE X.

CONTENTS.—Examples of the Quantity of Water required for Condensation of Steam with a Jet Condenser continued—Also with a Surface Condenser—Tube Surface required under different conditions.—Questions.

NOW, let us try the same two questions given in the last Lecture, but on the Cent. scale, if for no other purpose than to observe the great advantage of using that scale.

Either referring to our comparative table of thermometer scales, or by calculation (Lecture III.), we observe that—

212° F. corresponds to 100° C.
60° F. „ 15°·5 C.
100° F. „ 37°·7 C.
and 1146 units on F. scale = 637 on C. scale.

∴ Taking the first example of 1 lb. of water at 100° C., mixed with x lbs. of water at 15°·5 C., the resulting temperature being 37°·7 C., &c.

The Loss of Heat from the Water at 100° *C.* = *the Gain of Heat by the Water at* 15°·5 *C.*

$$\therefore 1 \times (100^\circ - 37^\circ{\cdot}7) = x \times (37^\circ{\cdot}7 - 15^\circ{\cdot}5),$$

$$\therefore 62{\cdot}3 = 22{\cdot}2\,x,$$

$$\therefore x = \frac{62{\cdot}3}{22{\cdot}2} = 2{\cdot}8 \text{ lbs.}$$

Again, taking the second example of 1 lb. of steam at 100° C., mixed with x lbs. of water at 15°·5 C., the resulting temperature being as before 37°·7 C.

The Loss of Heat from the Steam at 100° *C.* = *the Gain of Heat by the Water at* 15°·5 *C.*

$$\therefore 1 \times (637 - 37^\circ{\cdot}7) = x \times (37^\circ{\cdot}7 - 15^\circ{\cdot}5),$$

$$\therefore 599{\cdot}3 = 22{\cdot}2\,x,$$

$$\therefore x = \frac{599{\cdot}3}{22{\cdot}2} = 26{\cdot}9 \text{ lbs.}$$

If we had taken round numbers on the Cent. scale, as we did on the Fah. the advantage would have been still more apparent. We give another example on the subject of our last lecture more in accordance with what takes place in actual practice. The points we have hitherto considered were meant to lead up to this one. We shall again refer to this question of the

quantity of water required for condensation when we come to compare the relative efficiences of jet and surface condensers.

EXAMPLE.—A vacuum gauge placed in the exhaust pipe of a low-pressure cylinder indicates 26 inches, while the mercurial barometer stands at 30 inches. The temperature of the hot-well is 100° F., what is the minimum weight of injection water at 60° F. that will produce this result per pound of steam entering the condenser?

Let 30 in. by barometer correspond to 15 lbs. absolute;

Then, $$30 \text{ in.} : 26 \text{ in.} :: 15 : y$$

$$y = \frac{15 \times 26}{} = 13 \text{ lbs. } \textit{vacuum.}$$

Therefore, a 26-inch vacuum corresponds to a back pressure of $(15 - 13) = 2$ lbs. absolute. Now, 2 lbs. absolute pressure corresponds closely to a temperature of 126° F. $= t_1$ (see p. 74).

The Loss of Heat from the Steam at 126° *F.* = *the Gain of Heat by the Water at* 60° *F.*

The 1 lb. of steam at 126° F. *loses* $1 \times \{1082{\cdot}4 + {\cdot}3\, t_1 - (100^\circ - 32^\circ)\}$,
$= 1052{\cdot}2$ B.T.U.

The x lbs. of water at 60° F. *gains* $x \times (100^\circ - 60^\circ)$,
$= 40\,x$ B.T.U.

$$\therefore\ 1052{\cdot}2 = 40\,x,$$

$$\therefore\ x = \frac{1052{\cdot}2}{40} \quad 26{\cdot}3 \text{ lbs.}$$

It will thus be clear on comparing this result with the last example in Lecture IX., that almost the same weight of water is required per pound of steam, whether the steam exhausts at atmospheric pressure or not, for it is the 966 units of *latent heat* which the injection water has to contend with, more than the few units of *sensible heat* in the steam.

We observe that the hot well was 100° F. This gives off steam vapour corresponding (see Regnault's tables, Lecture VII.) to an absolute pressure of ·942 lb. on the square inch, or about equivalent to a 28 inch vacuum, supposing no air to be let free from the water. Of course, if air is set free, it will reduce the vacuum still further without a corresponding rise in temperature, and thus the advisableness, as we shall see further on,

of freeing from air as far as possible all feed-water to a boiler, especially when working with surface condensers.

In practice the hot-well water in sea-going steamers is kept at between 110° and 120° F. In order to do this, the cubic capacity of the jet condenser should not be less than one-third that of the cylinders exhausting into it, and the weight of injection water (at a velocity of 30 feet per second) in temperate climates from 25 to 30 times that of the steam, or from 30 to 35 times in the tropics. In the Red Sea the temperature of the sea water near surface in the summer season often exceeds 85° F.

The jet condenser has now, however, been almost entirely superseded by the surface condenser, owing to its many important advantages, which will be fully detailed when we come to describe marine engines using high-pressure steam. Owing to the fact that the condensing water does not come into direct contact with the exhaust steam, the temperature of the former is not raised quite so much as with the jet condenser, in practice it is probably never raised much more than 40° F., so that a slightly larger quantity of it is required. The condensing water is forced by means of a circulating pump through a double or treble series of brass tubes about ¾ inch external diameter, and generally ·048 inch thick. Usually the water is sent first through the lower tier of tubes, and then through the upper, thereafter discharging freely through the ship's side into the sea, so that the exhaust steam impinges against the sides of the warmer or upper set of tubes.

Suppose the exhaust steam to enter the condenser at a mean absolute pressure of 3 lbs., corresponding to a temperature of 142° F. (t_1), and to be condensed into water at 120° F. (t_2); also, that the circulating water enters at 60° F. (t_3), and is discharged at 100° F. (t_4), how many pounds of circulating water will be required per pound of steam?

The Total Loss of Heat from Steam = the Total Gain by Water.

$$1 \text{ lb. } \{(1082{\cdot}4 + {\cdot}3t_1) - (t_2 - 32)\} = x \text{ lbs. } (t_4 - t_3).$$

$$1\{(1082{\cdot}4 + {\cdot}3 \times 142) - (120 - 32)\} = x(100 - 60).$$

$$1037 = 40x; \therefore x = 25{\cdot}9 \text{ lbs.}$$

So we see that with the least practicable loss in heat of the steam and the highest desirable gain of heat by the water, not less than 26 lbs. of water are required with the surface condenser, whereas with the same temperature loss in steam and rise of water temperature from 60° to 120°, the theoretical quantity of water required with the jet condenser would only have been about 17 lbs. It is usual to allow about 40 times the weight of steam for general traders and 50 times for ships always in the tropics.

Size of Surface Condensers.—The foregoing calculations are simple,

interesting and instructive exercises for the student. They do not tell him the size of condensers adopted in practice. When ordinary low-pressure condensing engines were used, it was customary to specify for so many sq. ft. of condenser surface per indicated horse-power, exhausting at such and such a pressure—*e.g.*, "with a terminal pressure of 6 lbs. absolute, use 1·5 sq. ft. per I.H.P." Now, however, since the introduction of compound and multiple expansion engines, such a "rule of thumb" does not hold good, for it is evident that the weight of steam used and to be condensed, varies considerably for a given horse-power with the initial pressure. For example, two different engines indicating the *same power* and having the *same terminal pressure* may use the one steam of 60 lbs., and the other of 200 lbs. initial pressure; consequently, the weight of steam to be condensed is much less in the second case than in the first, and, therefore, a less condenser surface would suffice for it. A common rule is that of specifying a certain condenser surface per sq. ft. of boiler heating surface, and the author finds that one eminent firm adopts an average of ·7 sq. ft. per sq ft. of boiler heating surface with natural draught. In order, however, to place this matter more thoroughly before the student a table marked "Surface Condensers" has been added to this lecture, from which it will be seen that from 22 examples the most natural rule seems to be *the ratio subsisting between the condenser surface, and the product of the capacity of low-pressure cylinder with the terminal pressure*, which gives a mean of 3·24. It is very seldom that engineers go to the trouble of measuring either the weight of circulating water or its rise in temperature.

MEAN OF TWENTY-TWO EXAMPLES OF SURFACE CONDENSERS, MADE AND FITTED ON THE CLYDE BETWEEN 1875 AND 1890.

Terminal Pressure in Lbs. per Square Inch.	Indicated Horse-Power.	Revolutions per Minute.	Volume of Low-Pressure Cylinder in Cubic Feet.	Capacity of Double-Acting Circulating Pump in Cubic Feet.	Condensing Surface in Square Feet.	$\frac{\text{Condensing Surface}}{\text{I.H.P.}}$	Boiler Heating Surface in Square Feet.	$\frac{\text{Condensing Surface}}{\text{Capacity of L.-P. Cylinder} \times \text{Terminal Pressure}}$	$\frac{\text{Condensing Surface}}{\text{Heating Surface}}$
10	517	83	36·37	·85	1,024	1·95	1,793	3·24	·553

Here it will be observed, that with a mean terminal pressure in the low-pressure cylinder of 10 lbs. absolute (or a very low vacuum of not more than 10 inches) we have 1·95 square feet of cooling surface in the condenser for every I.H.P., whereas, with the British battleships, cruisers, sloops, and gunboats, built between 1900 and 1903, the mean is 1·1 square feet per I.H.P. In the U.S. of A. recent similar men-of-war give a mean of 1·3 square feet per I.H.P., and for torpedo-boat destroyers the ratio is only ·8 square foot per I.H.P. This shows the decided tendency to cut down the size and weight of surface condensers. By using more rapidly-acting centrifugal circulating pumps for circulating the water through the condenser, and also from the fact, that with higher pressures and triple-expansion engines a greater I.H.P. is obtained for the same weight of steam used. This may be still further reduced in future by employing highly superheated steam.

LECTURE X.—QUESTIONS.

1. What quantity of water is required to obtain one cubic foot of steam at 212° F.? What quantity of heat exists in such steam without being sensible to the thermometer? How much water at 60° F. should you allow for the condensation of each cubic foot of steam at 212° F. during the working of an engine, hot-well 100° F.? *Ans.* 1·05 cb. in.; 36·7 units of heat; 1·02 lb.

2. The temperature of the hot well is maintained at 38° C., the temp. of the condensing water being 10° C. Find the amount of water for condensation per lb. of steam at atmospheric pressure. *Ans.* 21·4 lbs.

3. State the essential differences between jet and surface condensation of steam. Deduce a formula for determining the weight of condensing water required in a surface condenser, in order to condense steam at a given temperature into water at a given temperature. The vacuum gauge of a surface condenser indicates 22 inches while the mercurial barometer stands at 30 inches. The temperature of hot well = 110° F. The condensing water enters at 60° F. and is discharged at 90° F. The weight of steam passing through the condenser per minute = 80 lbs. Find weight of condensing water required per hour. *Ans.* 168,048 lbs. = 75·02 tons.

4. Suppose that in a jet and in a surface condenser the temperature of the exhaust steam is the same (say 150° F.), also the temperature of the hot-well water (say 120° F.), and the condensing water (say 60° F.) If the circulating water in the latter only rises 30° F., what are the relative amounts of water required per lb. of steam? *Ans.* 17·3 lbs. and 34·6, or as 1 to 2.

5. In a surface condenser the tubes are $\frac{3}{4}$ inch outside diameter, 6 feet long and ·05 inch thick. How many such tubes will be required, and what will be the total cooling surface in square feet supposing the terminal pressure of exhaust to be 6 lbs., and the I.H.P. 1000? Again, suppose the engine to require $\frac{1}{2}$ lb. of steam per I.H.P. per minute, and the conditions as to temperatures to be the same as in the last question, how many pounds or cubic feet of circulating water will have to pass through the condenser per minute? *Ans.* 1,273 tubes; 1,500 sq. ft.; 17,320 lbs.; 277 cb. ft.

6. In an ordinary good surface condenser 16 lbs. of steam at 55° C. condenses per hour per square foot of surface, the temperature of the circulating water being 15° C.; how much heat per second is this per square centimetre of surface? If the two skins of the brass were at 55° C. and 15° C., the brass being 0·05 inch thick, the conductivity of brass in C.G.S. units being 0·2, find the heat which would pass per square centimetre. To what do you ascribe the small flow of heat in the surface condenser? Is there any remedy? (B. of E., 1901, H., Part i.)

7. In an engine trial the hot well discharge per minute was 29·7 lbs., the initial and final temperatures of the circulating water were 43·6° F. and 96·7° F. respectively, the temperature of the condenser steam was 123·7° F. and the temperature of the hot well was 113° F. Assuming the condenser steam to be just dry, find the number of lbs. of circulating water per minute. (C. & G., 1903, O., Sect. C.)

LECTURE XI.*

CONTENTS.—Work Done during the Conversion of Water into Dry Steam—Definitions of Internal and External Work—Efficiency of Steam—Efficiency of High Pressure Steam—General Expressions for External and Internal Work during Evaporation—Example I.—Heat Rejected to Condenser—Example II.—Partial Evaporation—Example III.—Generation of Steam in a Closed Vessel—Factor of Evaporation—Examples IV. and V.—Steam Calorimeter or Dryness Fraction Indicator—Examples VI. and VII.—Questions.

Work Done during the Conversion of Water into Dry Steam.—We can now give a more definite account of the distribution of heat expended during the conversion of water into steam, and thus prepare the way for a more thorough understanding of the economical use of steam in a steam engine.

An ordinary steam engine consists essentially of—

1. A *boiler* wherein the steam at a given pressure is generated from water at a given temperature.

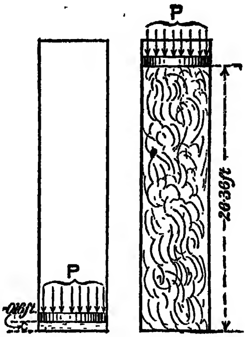
2. A *cylinder* containing a movable, steam-tight piston, on which the steam acts and does useful work.

3. Frequently, another part, called the *condenser*, is added. The function of the condenser is exactly the opposite of that of the boiler. For in it, the steam is converted back again into water after passing through the working cylinder. Engines having only the first two essential parts are called *non-condensing*, whilst those consisting of the three parts are called *condensing* engines. These three organs are usually quite distinct and separate from each other, the connections being made by pipes, valves, &c. For our present purposes it will be best to leave out of account all connections such as pipes and valves. We shall therefore suppose the boiler, working cylinder, and condenser, to be one and the same vessel. Also, we shall neglect all losses of heat, such as that due to radiation, conduction, &c. Further, we shall, in the meantime, consider the case of 1 lb. of water at an initial temperature of 32° F., raised into *dry* steam at 212° F. The pressure of the steam is, therefore, that due to atmospheric pressure—viz., about 14·7 lbs. per square inch.

Take a tall cylindrical vessel fitted with a weightless and perfectly frictionless piston, and place between the piston and the bottom of the vessel 1 lb. of water at 32° F. The cylinder being

* Students will observe that some of the values quoted in this Lecture, up to p. 135, from the new Table II., do not exactly agree therewith. These numbers are from the Steam Table in previous editions, but as they only vary from the more exact new table by a small amount they do not affect the principles and final values materially.

open at the top the pressure on the piston will be constantly that due to the atmosphere. For convenience, suppose the cross



ILLUSTRATING EXTERNAL WORK DONE DURING EVAPORATION OF 1 LB. OF WATER FROM AND AT 212° F.

sectional area of the area of the cylinder to be *one square foot* (or 144 square inches). Then,

$$\textit{Total pressure on piston} = P = 144 \times 14{\cdot}7 = 2116{\cdot}8 \text{ lbs.}$$

Since 62·5 lbs. of fresh water occupy a volume of 1 cubic foot,

$\therefore$ 1 lb. „ „ occupies „ $\frac{1}{62{\cdot}5} = {\cdot}016$ cub. ft.

The cross area of the cylinder being 1 square foot, it follows that the under surface of the piston will be ·016 foot above the base of the vessel.

By applying heat to the bottom of the vessel the temperature of the water will be ultimately raised to 212° F. The heat expended in this operation is $(212 - 32) = 180$ B. T. U. Now, the volume of the 1 lb. of water at the end of this operation is slightly greater than ·016 cubic foot, as shown by the graphic figure on page 67. The piston has, therefore, been raised by a small amount, and consequently work has been done in overcoming the atmospheric resistance. We thus see that rather *less* than 180 B. T. U. are employed in increasing the molecular kinetic energy of the water. This increase in the volume of the water between 32° F. and 212° F. is so small (being only $\cdot 016 \times \cdot 043 = \cdot 000688$ cubic foot (see Fig., page 104)* that it may

* The volume at 32° F. of a certain quantity of water is (as shown by the figure and text at page 108) = 1·000127, and at 212° F. = 1·043. The difference is practically = ·043. Consequently if a certain weight of water occupies about unit volume at 32° F. and increases by ·043 unit when its

safely be neglected. The piston therefore remains stationary between those two temperatures.

Continuing the application of heat to the water at 212° F., the same becomes evaporated and the piston rises rapidly, whilst the temperature remains constant. Suppose the source of heat to be withdrawn just when the last particle of the 1 lb. of water has been converted into dry steam. Then we know that 966·6 B. T. U. have been spent in bringing about this change. The piston will now be at a considerable height above the base of the vessel, and, consequently, a certain fraction of the *latent* heat will have been employed in doing work against atmospheric pressure. Referring to column V_s of the "Table of the Properties of Saturated Steam," Lecture VII, we notice that 1 lb. of *dry* steam at atmospheric pressure (temperature 212° F) occupies a volume of 26·36 cubic feet. Hence the piston will now stand at a height of 26·36 feet above the base of the vessel. The vertical displacement of the piston is, therefore, 26·36 − ·016 = 26·35 feet approximately.

$$\therefore \quad \textit{Work done in raising piston} = 2{,}116{\cdot}8 \times 26{\cdot}35 \text{ ft. lbs.}$$

$$\quad\;\;\text{,,} \qquad\quad \text{,,} \qquad\quad \text{,,} \qquad = 55{,}777\ 68 \qquad \text{,,}$$

$$\text{Or, expressed in heat units} \quad \text{,,} \quad = \frac{55{,}777\ 68}{772} = 72{\cdot}25 \text{ B.T.U.}$$

Thus, of the 966·6 B. T. U. of *latent* heat, 72·25 B. T. U. are employed in doing mechanical work *external* to the substance (water) which is undergoing a change of state; while the remainder (894·35 B. T. U.) is spent in bringing about *internal* changes.

DEFINITION.—*The energy spent in bringing about internal or molecular changes in a substance is called* **Internal Work,** *and that spent on bodies external to the substance is called* **External Work.**

The student must carefully distinguish between *internal* and *external* work. The former represents energy *in* the substance itself, whether in the form of molecular kinetic energy or that due to change of state; the latter represents energy which has *passed out of* the substance to external bodies.

temperature is raised to 212° F., what will be the increase in volume of ·016 cubic foot of water under the same circumstances? *Ans.* Less.

$$\therefore \quad 1 : {\cdot}016 :: {\cdot}0143 : x.$$

$$\textit{Or,} \qquad x = \frac{{\cdot}016 \times {\cdot}0143}{1} = {\cdot}000688.$$

c O
affect

The distribution of heat in converting 1 lb. of water at 32° F. into dry steam at 212° F., may be briefly stated thus—

1. *Raising temp. of water from 32° F. to 212° F.*	=	180·00 B.T.U.
2. *Internal work during evaporation* . .	=	894·35 "
3. *External work during evaporation* . .	=	72·25 "
Total Heat Expended . .	=	1146·6 B.T.U.

These numbers are in the proportion—180 : 894·35 : 72·25. Or, dividing by the smallest number, 72·25, the proportion is 2·5 : 12·38 : 1. We shall make use of the terms of this proportion in setting out the diagrams of work in the case under consideration.

The student knows from his study of mechanics that mechanical work can be completely represented by an area or "*diagram of work.*" When the effort or pressure is constant throughout the displacement (as in the case of the rising piston just referred to), the *diagram of work* is a rectangle, whose height represents the constant pressure, and base the given displacement. If the pressure varies during the displacement (as in the case of steam or gas expanding behind the working piston of an engine), the diagram of work will not be a rectangle, but a figure bounded by straight and curved lines. In this case, the *mean height* of the figure is a measure of the *mean pressure* exerted during the total displacement, and the length of the figure as before represents the total displacement.*

Now, heat and work being mutually convertible, it follows that quantities of heat may just as conveniently be represented by areas as quantities of mechanical work. These quantities, however, differ in this respect. In the former there is nothing corresponding to the two factors, effort or pressure and displacement, as in the case of the latter. Hence the diagram for a quantity of heat may be any shape we please, so long as it contains as many units of area as there are units of heat to be represented. It is, however, convenient for our present purposes to represent quantities of heat by rectangular areas, and if we first draw an ordinary diagram for the *external* work done during evaporation, we may then construct the *internal* work diagrams on the same base, the heights of which need only be drawn in the

* For further information and examples on the subject of graphical representation of work done by constant and by variable forces, see Lectures I. and II. of the Author's "Manual on Applied Mechanics." See also Lectures XIII., XIV., XVII., and XVIII., of the present work for theoretical and actual indicator diagrams of work.

proportions stated above. This should be clearly understood from what follows.

We have seen that the expression for the external work is the product of the two factors—viz., *pressure* = 2116·8 lbs., and *displacement* = 26·35 feet.

Or, *External work* = 2116·8 × 26·35 = 55,777·68 ft. lbs.

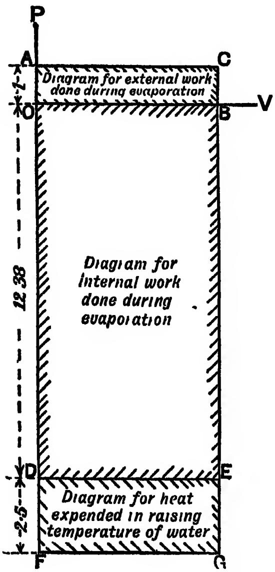


DIAGRAM SHOWING INTERNAL AND EXTERNAL WORK DONE IN CONVERTING WATER AT 32° F. INTO DRY STEAM AT 212° F.

Draw two lines O P, O V at right angles to each other. Along O P set off O A, to any convenient scale, to represent the pressure of 2116 8 lbs.; and along O V set off O B, to any convenient scale, to represent the displacement of 26·36 ft. Complete the rectangle O A C B. Then O A C B is the diagram representing the *external work* done during the evaporation of 1 lb. of water from and at 212° F. For its area is equal to O A × O B, which is thus proportional to 2116·8 × 26·36, or 55,777·68, the number expressing the ft.-lbs. of external work done.

Now, we have seen that the *internal* work done during evaporation is 12·38 times the external work. Therefore, produce P O downwards, and cut off a part O D = 12·38 × O A, and complete the rectangle O B E D. Then the area, O B E D, represents to the same scale as in the previous case, the *internal* work done during the evaporation of the water.

Similarly, on O D produced, cut off D F = 2·5 × O A, and complete the rectangle D E G F. Then the area, D E G F, represents the work done in raising the temperature of the water from 32° F. to 212° F.

Efficiency of Steam.—By returning the whole of the steam to its initial conditions—viz., water at 32° F., with the piston ·016 feet above the base of the cylinder, and repeating the above cycle of operations (heating, evaporating, condensing and cooling) over and over again, the piston will have a vertical reciprocating motion corresponding to that of an ordinary steam-engine. The *maximum external work* done during each cycle will be repre-

sented by the small rectangular area O A C B, while the total heat expended will be represented by the much larger area, A C G F. From this we can deduce the expression for the efficiency of a non-expansive engine using steam at atmospheric pressure from feed water at 32° F.* Thus—

$$\textbf{Efficiency} = \frac{\textbf{Heat converted into useful work.}}{\textbf{Total heat expended.}}$$

$$\text{,,} \quad = \frac{72\cdot25}{1146\cdot6} = \cdot063, \textit{ or}, 6\cdot3\,\%$$

Hence, under circumstances more favourable than any occurring in practice, we see what a small percentage of the total heat expended can be usefully employed in the engine.

The efficiency just found is usually called the **Steam Efficiency**, to distinguish it from the efficiencies of the boiler and the mechanism of the engine. The *product* of the efficiencies of the boiler and the engine constitutes the efficiency of the whole combination.

By using feed water at a higher temperature than 32° F., the total heat expended per 1 lb. of water evaporated would be less than that found above, and, consequently, the steam efficiency would be slightly higher. Thus, in jet-condensing engines, the feed water has a temperature corresponding to that of the hot well; which, in the average, is about 110° F. Hence, taking steam at atmospheric pressure (as before) raised from feed water at 110° F., we may calculate the steam efficiency as follows:—

Total heat expended = *Increase of Sensible heat* + *Latent heat.*

,, ,, = $(212 - 110) + 966\cdot6 = 1068\cdot6$ B.T.U.

External work done = $72\cdot25$ B.T.U. (same as before).

$$\therefore \textbf{ Steam Efficiency} = \frac{72\cdot25}{1068\cdot6} = \cdot0676, \textit{ or}, 6\cdot76\,\%.$$

This gives an increase of ·46 % over the first case.

Efficiency of High Pressure Steam.—Suppose we load the piston of the tall cylindrical vessel to such an extent that the pressure produced on the surface of the 1 lb. of water is, say, 100 lbs. per square inch absolute. From what has been already said, we know that steam will not begin to be formed (*i.e.*, the water will not boil) until the temperature is considerably higher than 212° F. The exact temperature at which evaporation commences can be found from the Table in Lecture VII. Referring to this Table we see in columns 1 and 2 that the boiling point of water subjected

*For the more advanced problem of finding the efficiency of an expansive engine, see Prof. Cotterill's book, "The Steam Engine as a Heat Engine."

to a pressure of 100 lbs. per square inch is 327·58° F., say, 328° F. To make the problem before us more practical, suppose the temperature of the 1 lb. of water to be 110° F.

Applying heat to the bottom of the vessel the temperature of the water rises to 328° F., at which point it remains fixed until evaporation is complete. During evaporation the piston ascends as before, but not to the same height. Referring again to Table II., Lecture VII., we notice, in column V_S, that the volume of 1 lb. of dry steam at a pressure of 100 lbs. per square inch is 4·403 cubic feet. Hence, after complete evaporation, the piston will be at a height of 4·403 feet above the base of the vessel. The total pressure on the piston is $P = 144 \times 100$ lbs.

$$\therefore \text{ External work, } H_E, \text{ during evaporation per lb. of water, } \left\{ \begin{array}{l} = P \times V_E = P(V_s - V_w) \\ = (144 \times 100) \times (4{\cdot}403 - {\cdot}016) \text{ ft.-lbs.} \\ = 63{,}173 \text{ ft.-lbs.} \end{array} \right.$$

$$\text{Or,} \qquad H_E = \frac{63{,}173}{778} = 81{\cdot}19 \text{ B.T.U.}$$

$$\textit{Total heat expended} = \left\{ \textit{Increase of Sensible heat} + \textit{Latent heat.} \right.$$

$$\textit{Increase of Sensible heat} = 328 - 110 = 218 \text{ B.T.U.}$$

$$\textit{Latent heat of steam at } 328^\circ \text{ F.} = 966 - {\cdot}7\,(328 - 212) = 884 \text{ B.T.U.}$$

$$\therefore \text{ Total Heat Expended} = 218 + 884 = 1{,}102 \text{ B.T.U.}$$

$$\therefore \qquad \text{Steam Efficiency} = \frac{81{\cdot}19}{1{,}102} = {\cdot}0736, \textit{ or, } 7{\cdot}36 \text{ per cent}$$

Comparing these results with the corresponding ones for steam at atmospheric pressure, we notice that the external work in this case is only 8·94 B.T.U. more than in the former case. This corresponds to an increase of about 13·3 per cent. The increase in the steam efficiency, however, is but 7·36 – 6·76, or ·6 per cent.

The student may therefore naturally ask, wherein lies the advantage of using high-pressure steam? In answer to this question, we should first of all remind him that the engine under consideration is a *non-expansive* one. That is, the steam acts on the piston with its full pressure throughout the whole stroke. Consequently, high-pressure steam would not be adopted except as a means of increasing the power of such an engine without increasing its size. For, the use of high pressure necessitates the employment of stronger boilers and cylinders, as well as greater accuracy in construction. It is only where steam is used expansively that high pressures can be economically and efficiently adopted.

Note.—Joule's equivalent has been taken as 778 in the above, for it is the most recent value, as noted at end of Lecture VI.

In drawing the above comparison between the performances of the two engines (the one using low-pressure and the other high-pressure steam) we have taken equal *weights* of steam. The results of the comparison would, however, be very different if we had taken equal *volumes*. Thus, it is quite clear that steam at 100 lbs. pressure, when used non-expansively in a cylinder of given volume, would perform $\frac{100}{15} = 6 \cdot \dot{6}$ times more work than steam at atmospheric pressure under like circumstances in the same cylinder. But, then, the *weights* of steam used in the two cases would be very nearly in the proportion $6 \cdot \dot{6} : 1$, and the fuel consumed would be in the same proportion. Now, the object of the engineer is to obtain the greatest amount of work for the least possible consumption of fuel, and, consequently, the comparison between the performances of two engines should be made with respect to the weights of steam used for a given amount of work performed. Nevertheless, it is sometimes necessary to know the work done per cubic foot of steam used. This may be obtained by dividing the work done per lb. of steam by the volume of 1 lb. of steam at the given pressure. Thus—

$$\left.\begin{matrix}\textit{Work done per cub. ft. of}\\ \textit{steam at atmos. pressure.}\end{matrix}\right\} = \frac{\textit{External work during evaporation.}}{\textit{Volume of 1 lb. of steam.}}$$

$$\text{,,} \qquad \text{,,} \qquad = \frac{55{,}777 \cdot 68}{26 \cdot 36} = 2116 \text{ ft. lbs.}$$

General Expressions for External and Internal Work during Evaporation.—We shall now express the preceding results in general terms—

Let t_1 = Temperature of steam.
,, t_2 = Temperature of feed water.
,, L = Latent heat at temperature t_1.
,, p = Pressure of steam in lbs. per square inch.
,, V_s = Volume in cub. ft. of 1 lb. of *dry* steam at pressure p.*
,, V_w = ,, ,, ,, water = ·016 cub. ft.

Supposing the steam to be *dry*, then, we have—

1\. Total heat expended = $\begin{cases}\textit{Increase of Sensible heat}\\ \textit{+ Latent heat.} \quad (\textit{See} \text{ LECTURE IX.})\end{cases}$

$$\text{,,} \qquad \text{,,} \qquad = (t_1 - t_2) + \text{L},$$
$$\text{,,} \qquad \text{,,} \qquad = (t_1 - t_2) + 966 \cdot 6 - \cdot 7\,(t_1 - 212) \text{ B.T.U.}$$
$$\text{,,} \qquad \text{,,} \qquad = 1{,}115 + \cdot 3\,(t_1 - t_2) \text{ B.T.U.}\dagger$$

* The volume of 1 lb. of dry steam at a given pressure is sometimes called the *Specific Volume* of steam at that pressure. We find, however, that students often make the mistake of confounding the term *Specific Volume* with that of "Relative Volume of Equal Weights of Steam and Water," and, therefore, we prefer not to use the former term.

† Instead of remembering this final result, students should deduce it, when required, from definition as stated in italics above.

2. **External work done during evaporation** $\Big\} = \Big\{$ *Pressure per sqr. ft. × Increase of volume during evaporation.*

" " $= 144\,p\,(V_s - V_w)$ ft. lbs.

Or, " " $= \dfrac{144\,p\,(V_s - V_w)}{772}$ B.T.U.

3. **Internal work done during evaporation.** $\Big\} =$ *Latent heat – External work.*

" " $= 966{\cdot}6 - {\cdot}7\,(t_1 - 212) - \dfrac{144\,p\,(V_s - V_w)}{772}$ B.T.U.

The value of V_w, in the expression for external work is so small compared with that of V_s for all ordinary pressures, that we may safely neglect it in most calculations.

*EXAMPLE I.—How many ft.-lbs. of work are done in converting 1 lb. of water from a temperature of 100° F. into dry steam at 281° F. (corresponding to an absolute pressure of 50 lbs. per square inch)? The volume of 1 lb. of dry steam at that temperature and pressure being 8·31 cubic feet; find external and internal work done during formation of steam, and weight of steam used per hour per horse power.

ANSWER.—Here, $t_1 = 281°$ F., $t_2 = 100°$ F., $p = 50$ lbs. per square inch, $V_s = 8{\cdot}31$ cubic feet.

1. *Total heat expended* $= \Big\{$ *Increase of sensible heat + Latent heat.*

Increase of sensible heat $= 281 - 100 = 181$ B.T.U.

Latent heat $= 966{\cdot}6 - {\cdot}7\,(281 - 212) = 918{\cdot}3$ B.T.U.

Total heat expended $= 181 + 918{\cdot}3 = \mathbf{1099{\cdot}3}$ **B.T.U.**

" " $= 1099{\cdot}3 \times 772 = \mathbf{848{,}659{\cdot}6}$ **ft. lbs.**

2. **External work done during evaporation** $\Big\} = 144\,p\,V_s.$

" " $= 144 \times 50 \times 8{\cdot}31 = \mathbf{59{,}832}$ **ft. lbs.**

3. **Internal work done during evaporation** $\Big\} = \Big\{$ *Latent heat of evaporation – External work.*

" " $= 918{\cdot}3 \times 772 - 59{,}832$ ft. lbs.

" " $= \mathbf{649{,}095{\cdot}6}$ **ft. lbs.**

4. Let x = Weight of steam used per hour per horse power.

External work done per 1 lb. of steam formed = 59,832 ft. lbs.

∴ " " " x lbs. " " $= 59{,}832\,x$ ft. lbs.

Now, 1 horse-power corresponds to $33{,}000 \times 60 = 1{,}980{,}000$ ft. lbs. per hour.

∴ By the conditions of the question—

$$59{,}832x = 1{,}980{,}000$$

Or,
$$x = \frac{1{,}980{,}000}{59{,}832} = \mathbf{33\ lbs.}$$

Heat Rejected to Condenser.—In the preceding examples it has been tacitly assumed that during the return motion of the piston within the cylinder, the condensation of the steam was effected under *zero* pressure—*i.e.*, condensation was so perfect that no *back pressure* was felt on the under surface of the piston. The piston, therefore, returned unloaded. The whole of the external

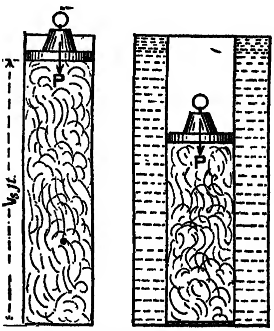
* Students should check this with p. 109 and new Table II.

work done during the upward motion of the piston was, therefore, available for useful purposes. Had the condensation been incomplete, part of the work would have been employed in returning the piston against the back pressure due to the imperfect vacuum. Such perfect conditions as we have hitherto assumed cannot be attained in practice. Condensation is always more or less imperfect, and consequently we find that the back pressure *varies* from 2 to 5 lbs. per square inch in condensing engines, to 15 or 18 lbs. per square inch in non-condensing engines. A perfect vacuum cannot be attained in practice; for, water at all temperatures gives off vapours which naturally exerts a certain pressure. Thus, at a temperature of about 80° F. water vapour exerts a pressure of about ·5 lb. per square inch, and at a temperature of 102° F. the vapour pressure is 1 lb. per square inch.

The subject presently before us is to determine the amount of heat rejected to the condensing water per lb. of steam passing through the engine. This, as may be inferred from the above remarks, depends upon the conditions under which condensation takes place. Consideration of the following three cases will give the student a clear idea of the distribution of heat in an ordinary steam engine:—

FIRST CASE.—*Suppose condensation to take place under the same pressure as the evaporation.*

Let p = Pressure of steam in lbs. per square inch absolute.
„ V_s = Volume of 1 lb. of dry steam at pressure p.
„ Q = Total heat expended per lb. from feed water temperature to steam at pressure p.
„ R = Rejected heat to condenser.



EXTERNAL WORK DONE DURING CONDENSATION OF STEAM UNDER THE SAME PRESSURE AS THE EVAPORATION TOOK PLACE.

As before, let the 1 lb. of water be heated under the movable piston of a tall cylindrical vessel whose cross sectional area is one square foot. For our present purposes, however, it is best to neglect the atmospheric pressure on the upper surface of the piston, and to suppose the necessary pressure to be caused by a weight placed on the piston. The magnitude of this weight will be, $P = 144\,p$ lbs.

After the water is completely evaporated, the piston will be at a height of V_s feet above the base of the vessel, and—

The external work done during evaporation $= PV_s$ ft. lbs.

$$" \quad " \quad " \quad " \quad = \frac{PV_s}{772} \text{ B.T.U.}$$

Suppose we now convert the tall cylinder into a condenser by surrounding it with cold water. The weight P, still remaining on the piston, condensation will take place under the same conditions that evaporation took place—viz., under a pressure of p lbs. per square inch. Let the final temperature of the condensed steam be the same as the initial temperature of the water. Then, during condensation, the heat rejected to the condensing water is clearly equal to the total heat expended, or Q units. For the heat rejected is derived from the following sources:

1. That heat which is derived from the work done by the descending piston. Neglecting the very small volume V_w of 1 lb. of water, we see that the work thus converted into heat is $\frac{P.V_s}{772}$ B.T.U., which passes through the steam into the condensing water.

2. The heat formerly spent on *internal work* during evaporation is now yielded up to the condensing water.

3. The *sensible heat* given out during the cooling of the water to its initial temperature.

Hence, in a cycle of operations of this kind, no *available* external work is done. The external work done during the ascent of the piston is undone, or has to be restored during its descent, thus leaving no work available for useful purposes.

The results of this case may be stated thus:

Heat Rejected to Condenser = Total Heat Expended.

Or, $R = Q.$

SECOND CASE.—*Suppose condensation to take place under zero pressure.*

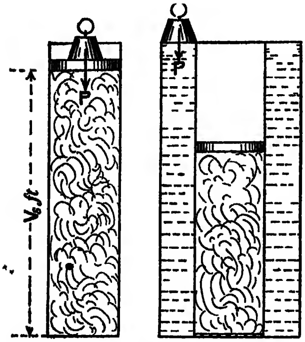
This corresponds to those cases previously considered, and also to the case of an engine whose condenser gives a perfect vacuum, or no back pressure.

To understand this case, suppose at the instant the condenser is applied to the cylinder full of dry steam that the weight [illegible] lifted [illegible] the piston. Then, clearly, no *external work* will [illegible] by

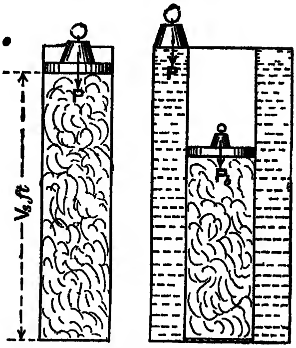
the descending piston during the condensation of the steam. Hence, in this case—

$$\textit{Heat rejected to Condenser} = \left\{\begin{array}{l}\textit{Total Heat Expended} \\ -\textit{External Work.}\end{array}\right.$$

Or, $$R = Q - \frac{PV_s}{772}$$



CONDENSATION OF STEAM AT ZERO PRESSURE.



CONDENSATION OF STEAM UNDER A BACK PRESSURE OF p_b LBS. PER SQUARE INCH.

THIRD CASE.—*Suppose condensation to take place under a pressure of p_b lbs. per square inch absolute.*

This corresponds to those cases occurring in practice, where p_b is the *back pressure* on the piston due to the pressure of the vapour in the condenser.

At the instant when condensation is about to take place, imagine the weight P to be lifted off the piston and another but smaller weight ($P_b = 144\,p_b$ lbs.) to be put in its place. Then, during condensation, the piston descends under the load P_b lbs., and the work done during the descent is converted into heat, which passes away to the condensing water. The heat rejected to the condenser is therefore greater in this than in the former case by the amount P_bV_s ft.-lbs., or $\frac{P_bV_s}{772}$ B.T.U.

$$\therefore\ R = Q - \frac{PV_s}{772} + \frac{P_bV_s}{772} \text{ B.T.U.}$$

$$\textit{i.e. } R = Q - \frac{(P - P_b)\,V_s}{772} \text{ B.T.U.}$$

Or, since $P = 144\,p$, and $P_b = 144\,p_b$,

$$R = Q - \frac{144}{772}(p - p_b)\,V_s \text{ B.T.U.}$$

The foregoing results could have been obtained at once from the *Principle of the Conservation of Energy*, thus—

Total heat expended = $\begin{cases}\textbf{Heat converted into useful work} \\ \textbf{+heat rejected to condenser.}\end{cases}$

But, *Total heat expended* = Q units.

Heat converted into useful work $= \frac{(P - P_b)\, V_s}{772} = \frac{144}{772}(p - p_b)\, V_s$ units.

Heat rejected to condenser = R units.

$$\therefore \qquad Q = \frac{144}{772}(p - p_b)\, V_s + R \text{ (heat units).}$$

As already explained, the back pressure in condensing engines varies from 2 lbs. to 5 lbs. per square inch. In non-condensing engines the steam, after performing work in the cylinder, exhausts into the atmosphere, and the back pressure can therefore never be less than the atmospheric pressure, in fact it varies from 15 to 18 lbs. per square inch. In this case the atmosphere is the condenser, but the whole of the heat rejected to it is lost. The advantages of a good condenser are thus apparent. For, in addition to the reduction of the back pressure, part of the heat rejected to it is employed in raising the temperature of the feed water.

EXAMPLE II.—A non-expansive condensing steam engine is supplied with steam at a pressure of 45 lbs. per square inch by gauge. The vacuum gauge indicates a pressure of 2 lbs. per square inch in the condenser. Find (1) the amount of heat rejected to the condenser per lb. of steam used; (2) the steam efficiency; and (3) the weight of water used per hour per effective horse-power. Let the temperature of feed water = 100° F.

ANSWER.—The *absolute* pressure of the above steam is 45 + 15 = 60 lbs. per square inch. Referring to Table II., Lecture VII., we find the temperature of steam at 60 lbs. absolute to be 292·51° F., say, 293° F.; and the volume of 1 lb. of dry steam at the same pressure is V_s = 7 cubic feet.

$\therefore$ *Total heat expended per lb. of steam* = *Increase of Sensible heat + Latent heat.*

Or, $Q = (293 - 100) + 966{\cdot}6 - {\cdot}7\,(293 - 212)$ B.T.U.

,, = 1103 B.T.U. (very nearly).

Heat converted into Useful Work per lb. of steam $= \frac{144}{772}(p - p_b)\, V_s$

,, ,, $= \frac{144}{772}(60 - 2) \times 7 = 75{\cdot}75$ B.T.U.

Now, $Q = \frac{144}{772}(p - p_b)\, V_s + R,$

$R = 1103 - 75{\cdot}75 = 1027{\cdot}25$ **B.T.U.**

Steam Efficiency $= \frac{75{\cdot}75}{1103} = {\cdot}0687$, *or*, **6·87%.**

Let x = weight of water used per hour per effective H.P.

Then, $\left.\begin{array}{r}\textit{Useful work done per}\\ \textit{x lbs. of water used}\end{array}\right\} = 144(p - p_b)V_s \times x$ ft.-lbs.

$$\text{,,} \quad \text{,,} \quad = 144(60-2) \times 7 \times x = 58{,}464\,x \text{ ft.-lbs.}$$

But, 1 *Horse-power* = $33{,}000 \times 60 = 1{,}980{,}000$ ft.-lbs. per hour.

$\therefore$ $58{,}464\,x = 1{,}980{,}000$.

$$\therefore \quad x = \frac{1{,}980{,}000}{58{,}464} = \mathbf{34{\cdot}4\ lbs.}$$

Partial Evaporation.—Up till now, our calculations, &c., have been based on the assumption that the steam when formed contained no suspended moisture. In other words, the steam was assumed to be perfectly *dry*. The steam supplied to an engine from an ordinary boiler is seldom in this condition, for it is always more or less saturated with watery particles. Even if the steam be dry on leaving the boiler, it may enter the working cylinder in a very moist condition, due to loss of heat from various causes in its passage from the former to the latter. Again, large quantities of water sometimes pass along with the steam from the boiler to the cylinder, and go through the engine without yielding full return for the heat spent in raising its temperature to that of the accompanying steam. Such large quantities of water are called *priming*, in distinction to the smaller quantities which are mingled with the steam in the form of a fine spray and which we have termed *suspended moisture*. Priming is generally the result of either too small a steam space in the boiler or too rapid ebullition, or impurities in the water, or a combination of these defects. It may cause a great deal of trouble to the engineer in charge, and when excessive, it may result in a fractured cylinder-cover or necessitate the stoppage of the engine.

At present we are, however, not concerned with the effects of priming, and shall consequently confine our remarks to cases of partial evaporation in which the steam contains moisture held in suspension.

Take the case of 1 lb. of water at 212° F. converted into *wet* steam at the same temperature. Suppose the steam contains 10 per cent. of suspended moisture. Then only 90 per cent., or ·9 lb. of the water will be in the form of *dry* steam. Hence, instead of spending the 966·6 B.T.U. of latent heat, we only require $\cdot 9 \times 966{\cdot}6 = 869{\cdot}94$ B.T.U. to bring about this result. The fraction, ·9, is called the *dryness fraction* of the steam. If the 1 lb. of water had had an initial temperature less than 212° F., say, 100° F., then, the total heat expended would have been $(212 - 100) + \cdot 9 \times 966{\cdot}6 = 981{\cdot}94$ B.T.U. Generally—

Let Q = Total heat expended per lb. of *wet* steam at temperature $t_1°$ from water at temperature $t_2°$.
" L = Latent heat per lb. of *dry* steam.
" x = Dryness fraction, or *dry* steam in 1 lb. of wet steam.

Then, Q = *Increase of sensible heat + latent heat.*

But, *Increase of sensible heat* $= t_1 - t_2$ *heat units*,

And, $\left.\begin{array}{l}\textit{Latent heat per lb. of}\\ \textit{wet steam formed}\end{array}\right\} = x\,L$, *heat units.*

Therefore, $$Q = (t_1 - t_2) + x\,L \text{ heat units.}$$

We have now to show how the *external work* done during the formation of *wet* steam is found.

Let V_s = Volume of 1 lb. of *dry* steam at pressure p lbs. per square inch.
V_{ws} = Volume of 1 lb. of *wet* steam at same pressure.
V_w = Volume of 1 lb. of water = ·016 cub. ft.
x = Dryness fraction (as before).

Then, V_{ws} = (*vol. of dry steam + vol. of water*) in 1 lb. of the mixture.
$$\text{"} \quad = x\,V_s + (V_w - xV_w),$$
Or, $$\left.\begin{aligned} V_{ws} &= xV_s + (1 - x)\,V_w \\ \text{"} &= x\,(V_s - V_w) + V_w \end{aligned}\right\}$$

Supposing, then, the piston of the cylinder to be one square foot in area, we get—

Displacement of piston $= V_{ws} - V_w$ ft.

$\therefore \left.\begin{array}{l}\textbf{External work per lb.}\\ \textbf{of wet steam formed.}\end{array}\right\} = 144\,p\,(V_{ws} - V_w)$

" " $= 144\,p\,x\,(V_{s} - V_w)$ **work units.**

Unless for very high pressures, V_w is very small compared with V_{ws}, and may, therefore, be neglected in the above formulæ.

EXAMPLE III.—A boiler supplies steam at a pressure of 90 lbs. absolute, which contains 10 per cent. of suspended moisture. The temperature of the feed water is 100° F. Find (1) volume per lb. of wet steam thus formed; (2) the external and internal work during evaporation; and (3) the total heat expended per lb. of steam used.

ANSWER.—Here, p = 90 lbs. abs., and temperature corresponding to this pressure is

$$t_1 = 320°\text{ F.}; \quad t_2 = 100°\text{ F.}; \quad x = \frac{100 - 10}{100}$$

Volume of 1 lb. of *dry* steam at pressure p is V_s = 4·79 cub. ft.
From above formulæ, we get—

1. **Volume of 1 lb. of wet steam.** $\Big\} = V_{ws} = x\,(V_s - V_w) + V_w$,

$= \cdot 9\,(4\cdot 79 - \cdot 016) + \cdot 016 = \mathbf{4\cdot 312}$ **cub. ft.**

2. **External work done per lb. of wet steam** $\Big\} = 144\,p\,(V_{ws} - V_w)$,

$= 144 \times 90\,(4\cdot 312 - \cdot 016)$ ft. lbs.

$= \mathbf{55{,}728}$ **ft. lbs.**

Or, " " $= \dfrac{55{,}728}{772} = \mathbf{72\cdot 2}$ **B.T.U.** *approximately.*

3. **Internal work done during evaporation.** $\Big\} = \Big\{$ *Latent heat per lb. of wet steam – external work.*

" " $= x\,L - \dfrac{144\,p\,(V_{ws} - V_w)}{772}$ heat units.

" " $= \cdot 9 \times \{966\cdot 6 - \cdot 7\,(320 - 212)\} - 72\cdot 2$ B.T.U.

" " $= 801\cdot 9 - 72\cdot 2 = \mathbf{729\cdot 7}$ **B.T.U.**

4. **Total heat expended per lb. of wet steam formed.** $\Big\} = \Big\{$ *Increase of sensible heat + latent heat.*

" " $= (t_1 - t_2) + x\,L$.

" " $= (320 - 100) + 801\cdot 9 = \mathbf{1022}$ **B.T.U.** *approximately.*

Generation of Steam in a Closed Vessel.—Having thus considered the whole process of the generation of steam under constant pressure, we shall now explain, briefly, the differences between those cases and the generation of steam in a closed vessel. This will be of interest to the student since it corresponds to the case of getting up steam in a boiler.

Suppose that we have 1 lb. of water at a given temperature enclosed in a vessel of large capacity. Suppose, further, that the only pressure on the surface of the water is that due to the pressure of its own vapour.

By applying heat to the bottom of the vessel, the temperature of the water rises as before, steam is generated and its pressure increases with its temperature. In previous cases, where the water was heated and evaporated under a loaded piston, no evaporation took place until the natural tension within the mass of water was sufficiently great to overcome the superincumbent pressure. In the present case, however, the surrounding pressure is always in equilibrium with the tension within the mass of water, and, consequently, evaporation goes on uninterrupted.

Suppose the capacity of the vessel to be 26·36 cubic feet (the volume occupied by 1 lb. of dry steam at atmospheric pressure). Then, when the temperature of the mass has risen to 212° F., the whole of the water will be converted into *dry* steam, and its pressure will be 14·7 lbs. per square inch absolute. Further application of heat causes superheating of the steam. Similarly, if the vessel had a capacity of 7 cubic feet (the volume occupied

by 1 lb. of dry steam at a pressure of 60 lbs. per square inch absolute), then complete evaporation would not occur until the temperature was 293° F., and the pressure of the 1 lb. of dry steam thus formed would be 60 lbs. per square inch absolute.

In getting up steam in an ordinary boiler, the pressure on the surface of the water at the commencement is usually equal to that of the atmosphere. On applying heat the temperature will rise, evaporation, or generation of steam, will not commence at once, but will be delayed until the temperature has risen to 212° F. after which the evaporation will proceed as described above.

We have seen that during evaporation under constant pressure, a fraction of the total heat expended is transformed into external work. But, by the nature of the present case, no such external work can be done, and this constitutes the essential difference between the two modes of forming steam. Now, it is quite impossible to conceive of any difference in the internal energy of 1 lb. of dry steam formed according to either method, so long as the pressures are equal. Hence we conclude, *that the total heat expended in evaporating water in a closed vessel is less, by the amount due to external work, than that spent in producing the same final result by evaporating under a constant pressure.*

It is true that during evaporation of the water in the closed vessel, work is being continually spent in compressing the steam already formed; this work, however, is done *within* the mass itself, and is but part of the *internal work.*

Equivalent Evaporation from and at 212° F. and Factors of Evaporation.—In comparing the evaporative results of different boilers, it is still a common practice to measure and to state their efficiencies by the weights of their feed waters converted into steam per pound of fuel burned per hour in their furnaces. This rough-and-ready method is open to several objections:—

1. The fuel may not have the same calorific value in each case.
2. The stoking may be skilfully performed in one case and not in another.
3. The temperatures of the feed water may be different.
4. The pressures of the steam may be different in each case.
5. One boiler may be producing *dry saturated steam*, whilst another may be giving off more or less *wet steam* and a third *superheated steam.*

It is therefore necessary, in making such tests, to fix upon a *fairer standard of comparison*, and the one which has found most favour hitherto with engineers, is called the "*equivalent evaporation from and at 212° F.*" *per lb. of fuel consumed per hour* And the *factor of evaporation* in this case is, the ratio of the weight of water which could be evaporated as dry steam from and at 212° F. to the weight of water which was actually heated up from the feed temperature to and evaporated at the pressure, temperature and dryness as steam in the boiler.

In Table II. under the symbol, E, the values are given for "Factor of Equivalent Evaporation at 212° F." for the various absolute pressures of

p lbs. per square inch, but these values are only applicable to a boiler feed water temperature of 212°, and simply mean that—

$$\left.\begin{array}{l}\text{The Factor of Equiv-}\\ \quad\text{alent Evaporation}\\ \quad\text{for Dry Saturated}\\ \quad\text{Steam in Table II.,}\end{array}\right\}E = \frac{\text{Weight of water which could be evaporated from and at } 212^\circ \text{ F. under atmospheric pressure}}{\text{Weight of water actually heated from } 212^\circ \text{ F. to and evaporated at pressures } p \text{ or temp. } t_e^\circ}$$

Let us first of all consider the case of dry saturated steam, or what is only too often assumed to be steam in that condition.

Let E_f = Evaporation factor where feed water is at t_f°.
H = Total heat, as in Table II., from water at 32° F. to temperature of evaporation t_e°.
W_e = Weight of steam per hour per lb. of fuel, at pressure p and temperature of evaporation t_e°.
W_a = Weight of steam per hour for same B.T.U. as with W_e from and at 212° F.

Then, by the above definition of this *standard of comparison*,

$$\text{The Factor of Evaporation, } E_f = \frac{W_a}{W_e}.$$

But, every 1 lb. of water evaporated from and at 212° F. has only to receive the latent heat of steam, or 966 B.T.U.

And, every 1 lb. of water raised from the feed temperature t_f° to the temperature of evaporation t_e° corresponding to the pressure p receives $H-(t_f^\circ-32^\circ)$ B.T.U.

Hence, $$W_a \times 966 = W_e[H-(t_f^\circ-32^\circ)].$$

Or, $$E_f = \frac{W_a}{W_e} = \frac{H-(t_f^\circ-32^\circ)}{966}.$$

And, this is what is meant by the "*Equivalent Evaporation from and at 212° F.*," as well as by the *Factor of Evaporation.*

EXAMPLE IV.—A boiler working at 100 lbs. absolute produces 10 lbs. of dry saturated steam per lb. of coal burned in its furnace, and when the temperature of the feed water is 100° F.; find the "equivalent evaporation from and at 212° F." and the "factor of evaporation." Referring to Table II. we see, that for p = 100 lbs., H = 1,181·9, and we are given t_f° = 100° F., and W_e = 10 lbs.

Hence, $E_f = \dfrac{H-(t_f^\circ-32^\circ)}{966}$, and, $W_a = E_f \times W_e$.

Or, $E_f = \dfrac{1,181\cdot9-(100-32)}{966}$;

$\therefore$ $E_f = \mathbf{1{\cdot}153}$; and, $W_a = 1\cdot153 \times 10 = \mathbf{11{\cdot}53\ lbs.}$

EXAMPLE V.—Suppose, that another boiler also working at 100 lbs. absolute produces 10 lbs. of *wet steam*, having 10 per cent of moisture for every 1 lb. of coal burned, when the temperature of feed water is also 100° F., find its factor of evaporation and equivalent evaporation from and at 212° F.

Here, everything is the same as in the previous example except, that we have a dryness fraction, x = ·9, since 10 per cent. or $\frac{1}{10}$ of the steam is wet, as we saw before, when considering partial evaporation in this lecture.

Hence, H is not the total heat of evaporation from 32°, as found in Table II., but $H = S-(t_f^\circ-32^\circ)+xL$; or the sensible heat plus latent heat per lb. of this wet steam.

Where, S = Sensible heat required to raise 1 lb. of water from $32°$ to $t_s°$.
or $S = 297·9$ B.T.U., from Table II.
And, L = Latent heat of dry steam at pressure p and temperature $t_s° = 884$, also from Table II.

Hence, $$E_f = \frac{S-(t_f°-32°)+xL}{966}; \quad \text{and, } W_a = E_f \times W_e.$$

Or, $$E_f = \frac{297·9-(100-32)+·9\times 884}{966};$$

$\therefore$ $E_f = 1·03$; and, $W_a = 1·03 \times 10 = 10·3$ lbs

Note.—If an engineer under these circumstances had assumed, that this boiler was generating dry saturated steam (instead of testing the same carefully for wetness by an instrument such as we are about to describe, and finding 10 per cent. moisture) he would have over-rated the boiler as capable of producing 11·53 lbs. of dry steam from and at 212° F. instead of only 10·3 lbs. This over-estimate would have been nearly 12 per cent. too much. We shall deal with superheated steam in Lecture XV.

Steam Calorimeter* or Dryness Fraction Indicator.—Of late, many efforts have been made to devise, construct, and use an instrument which would enable the engineer to tell accurately and quickly the wetness of the steam he was using under different circumstances and from different kinds of boilers. Although many forms of such an instrument have been placed at the disposal of the engineer, yet there seems to be a belief that they do not indicate correctly under widely different conditions. However, we herewith illustrate, describe, and give an example obtained by means of the steam calorimeter originally designed by Mr. George H. Barrus, of Boston, U.S.A., and made in this country by M'Innes-Dobbie, of Glasgow. This instrument, if thoroughly lagged throughout, and skilfully used does give useful results. Its construction and action will form a fitting termination to this Lecture, and the student should therefore study the same with due interest.

The following apparatus consists of two distinct parts, viz.:—

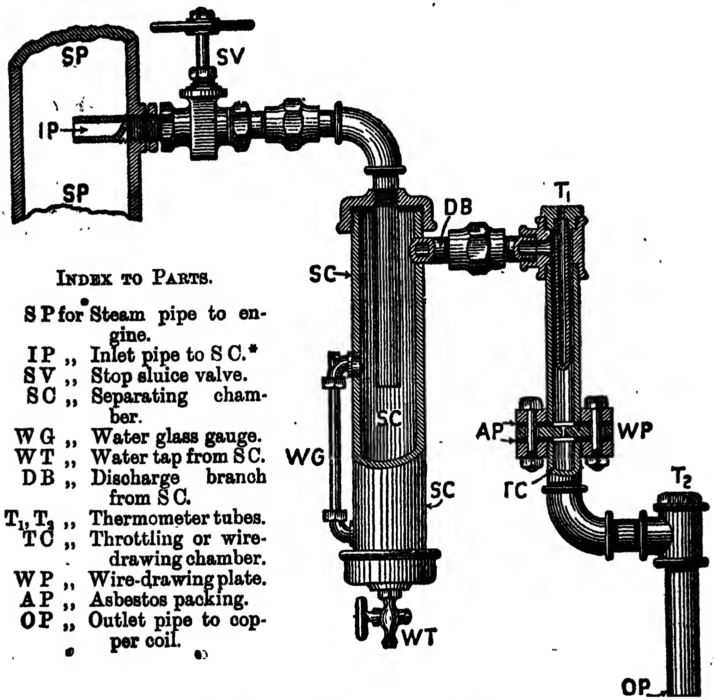
(1) The wet steam separator chamber, S C.
(2) The wire-drawing or throttling chamber, T C.

To enable the student to clearly understand the separate actions which take place in the two chambers, S C and T C, let us assume in the first instance that the throttling chamber is removed, and that the copper coil or cold water condensing pipe is transferred from the outlet pipe, O P, to the discharge branch D B. Then, the percentage weight of moisture which the

* As will be seen from the definition of calorimetry, and its derivation given in Lecture IV., the term steam calorimeter is a misnomer; for, we do not measure the *quantity of heat* per lb. of steam, but simply attempt to ascertain the *dryness fraction*, or the wetness or percentage of moisture in a steam supply from a boiler to a plant by means of this instrument.

separating chamber would trap and detect will be found as follows:—

To find the dryness fraction of steam by means of the separating chamber alone.—Steam must be allowed to pass for a short time



INDEX TO PARTS.

S P	for	Steam pipe to engine.
I P	,,	Inlet pipe to S C.*
S V	,,	Stop sluice valve.
S C	,,	Separating chamber.
W G	,,	Water glass gauge.
W T	,,	Water tap from S C.
D B	,,	Discharge branch from S C.
T_1, T_2	,,	Thermometer tubes.
T C	,,	Throttling or wire-drawing chamber.
W P	,,	Wire-drawing plate.
A P	,,	Asbestos packing.
O P	,,	Outlet pipe to copper coil.

THE BARRUS STEAM CALORIMETER.
Made by M'Innes-Dobbie, of Glasgow

* Some engineers object to the calorimeter steam inlet pipe, I P, having its end closed, with only thin slits on its top side, as this wire draws the steam at that point. It might, thereby, convey to the calorimeter drier steam than was passing along the steam pipe, S P, to the engines. This error may be obviated by so perforating I P that the steam freely enters it through holes drilled on all sides; the combined area of these holes being greater than the cross-sectional area of the I P pipe. The main object of the particular form of inlet to I P, as well as the distance to which it is inserted into S P, is to obtain a fair average sample of the steam which does pass, under recognised conditions, from the boiler to the engine or other appliance where it is used.

through the separating chamber, S C, by opening S V and W T in order to warm it; otherwise a quantity of water would accumulate, due to initial condensation. Then close W T, and the steam, after passing through the separating chamber, will be condensed by means of the copper coil which is attached to D B, and immersed in cold water. This coil should have sufficient cooling surface to condense all the steam. The condensed steam thus formed is collected in a vessel as it drops from the outer end of the copper coil, and carefully weighed.

When the copper coil is used for condensing the steam, the "dryness fraction" is calculated thus:—

Let W_1 = Weight of dry steam condensed by the copper coil.
W_2 = Weight of water found in the separating chamber, or weight of trapped moisture obtained from W T.
x = "Dryness fraction" of steam.

Then,

Total weight of steam used $= W_1 + W_2$. Hence, $x = \dfrac{W_1}{W_1 + W_2}$.

EXAMPLE VI.—If $W_1 = 2{\cdot}1$ lbs., and $W_2 = {\cdot}37$ lb.,

The *dryness fraction*, $x = \dfrac{W_1}{W_1 + W_2} = \dfrac{2{\cdot}1}{2{\cdot}1 + {\cdot}37} = {\cdot}85$.

Or, ,, ,, $x = 85$ per cent. of apparently dry steam.

The amount of moisture found in the steam is therefore about 15 per cent. by this first chamber alone.

Second.—To find the dryness fraction of steam by means of the wire-drawing chamber alone.—When the amount of moisture present in the steam is not above 3 to 4 per cent., a throttling calorimeter may be used. In this form of calorimeter the steam passes from the main steam pipe, S P, at boiler pressure into the throttling chamber, T C, wherein it falls nearly to atmospheric pressure, and passes away by the exhaust outlet pipe, O P, at the bottom of the chamber. The temperature, t_1, of the steam as it comes from S P, is obtained by a thermometer placed in the tube, T_1. This temperature, t_1, enables us to ascertain from the tables in Lecture VII. and text of Lecture IX. the sensible heat, S_1, of the steam at that temperature, t_1. The temperature, t_2, of the steam in this chamber, T C, is taken by means of a thermometer placed in tube T_2, and this temperature is then compared with the normal temperature, t_3, of the steam due to its pressure, as found by an attached steam gauge at this place, and from the tables in Lecture VII.

The action of this wire-drawing part depends upon the fact,

that the total heat of steam at the higher pressure is greater than the total heat at the lower pressure. Hence, a quantity of heat is set free from the steam as it drops in pressure. It is this heat which goes, *first*, to evaporate and convert the suspended globules of water into steam; and, *second*, to superheat the steam at the lower pressure, if the excess of heat be sufficient to do so.

Let H_{T_1} = Total *heat* per pound of steam passing T_1 at the temperature t_1.

„ H_{T_2} = Total heat per pound of steam passing T_2 due to its reduced pressure at temperature t_2.

„ S_1 = Sensible heat per pound of steam at temperature t_1 from 32° F.

„ S_2 = Sensible heat per pound of steam at temperature t_2 from 32° F.

„ L_1 = Latent heat per pound of steam at temperature t_1.

„ L_2 = „ „ „ t_2.

„ x = Dry steam per pound of steam generated, or the "dryness fraction."

„ H_σ = Specific heat of steam which is taken as ·48 and constant.

„ t_1 = Temperature of steam in main steam pipe, as measured at tube T_1.

„ t_2 = Temperature of steam below W P through which it has been wire-drawn to a lower pressure, and measured at tube T_2.

„ t_3 = Normal temperature of steam in T C due to its pressure by gauge.

But, $H_{T_1} = S_1 + xL_1$, and $H_{T_2} = S_2 + L_2$, when the moisture is just evaporated (from Lectures IX. and XI.).

If there be sufficient excess of heat to superheat the steam at T_2, then the heat required to do so $= H_\sigma\ (t_2 - t_3)$.

Therefore, $$H_{T_1} = H_{T_2} + H_\sigma\ (t_2 - t_3).$$

Or, $$S_1 + xL_1 = S_2 + L_2 + \cdot 48\ (t_2 - t_3).$$

Hence, $$x = \frac{S_2 - S_1 + L_2 + \cdot 48\ (t_2 - t_3)}{L_1}.$$

EXAMPLE VII.—Let t_1 = 338° F., t_2 = 250° F., and t_3 = 216·3° F. This last value is the temperature of saturated steam from table, Lecture VII., corresponding to an absolute pressure of 16 lbs. per square inch, or exhausting at, say, 1 lb. above atmospheric pressure.

Then, from the tables, Lecture VII., and reckoning from 32° F. as the zero of temperature, we get:—

$$H_{T_2} = 1{,}147·9 \text{ B.T.U.},\ L_1 = 876·3 \text{ B.T.U.},\ S_1 = 308·7 \text{ B.T.U.}$$

Then, $$x = \frac{S_2 + L_2 + ·48\,(t_2 - t_3) - S_1}{L_1}.$$

Or, $$x = \frac{1{,}147·9 + ·48\,(250 - 216·3) - 308·7}{·876·3},$$

i.e., $$x = \frac{839·2 + 16·176}{876·3} = \frac{855·376}{876·3} = ·976.$$

Hence, x = 97·6 per cent. Or, the weight of moisture computed from the formula is about $2\frac{1}{2}$ per cent.

When the steam is first passed through the separating chamber, S C, and then through the throttling or wire-drawing chamber, T C, on its way to the copper pipe condensing coil, as in the form of instrument just described and illustrated, then the total percentage of moisture in the steam as it comes from the steam pipe, S P, is obtained by *one* test, and the separate results from the separator chamber, S C, and throttling chamber, T C, have to be added, to give the total moisture found in the steam.

In the above example we found that the water collected from the separating chamber, S C, was . . . = 15 per cent.
And, the water collected from the wire-drawing or throttling chamber, T C, was . = $2\frac{1}{2}$ „
Hence, the total quantity of moisture present in the steam as it came from S P was . = $17\frac{1}{2}$ „

Lecture XI.—Questions.

1. How many foot lbs. of work and units of heat are absorbed in converting 5 lbs. of water at 32° F. into *dry steam* at atmospheric pressure? Illustrate your answer by diagrams similar to that given in the Lecture, showing the internal and external work done on the water by the heat. *Ans.*

2. Define the terms "Internal Work" and "External Work," with reference to the generation of steam. How is the efficiency of a steam engine expressed? Illustrate your answers by taking an example and working out the various quantities arithmetically.

3. A boiler generates dry steam at an absolute pressure of 95 lbs. per square inch from feed water at 60° F. What percentage of heat will be saved by a feed-heater which raises the temperature of the feed water to 212° F.? *Ans.* 13·18 per cent.

4. A non-expansive engine uses steam at an absolute pressure of 60 lbs. per square inch, and makes 60 double strokes per minute. The area of the piston is 1 square foot, and the length of the stroke is 12 inches. Find (1) Weight of steam used per minute; and (2) Total heat expended per minute, the temperature of the feed water being as 60° F. *Ans.* (1) 17·14 lbs.; (2) 19,600 B. T. U.

5. A lb. of water at 60° F. is converted, at constant pressure, into dry steam at 75 lbs. per square inch absolute. Find (1) Total heat expended; (2) External work done during evaporation; (3) Internal work done during evaporation; (4) Work done in raising temperature of water. Construct a diagram showing graphically these various quantities of work. *Ans.* (1) 1148·35 B.T.U.; (2) 79·72 B.T.U.; (3) 821·13 B.T.U.; (4) 247·5 B.T.U.

6. Suppose, in Question 5, that the 1 lb. of water had been converted into wet steam containing 10 per cent. of suspended moisture. Find (1) Internal work; (2) External work done during evaporation. *Ans.* (1) 738·27 B.T.U.; (2) 71·73 B.T.U.

7. A boiler supplies steam with 10 per cent. of suspended moisture, the evaporation taking place at 320° F. from feed water at 100° F. Find total heat expended per 1 lb. of steam formed, and the weight of water which could be evaporated from and at 212° F. for the same expenditure of heat. *Ans.* 1022 B.T.U.; 1·056 lbs.

8. An engine works non-expansively with condensation. The initial pressure of the steam is 25 lbs. by gauge, and the back pressure is 3 lbs. absolute. Temperature of feed water 104° F. Find (1) Effective work per lb. of steam used; (2) Weight of steam used per hour per H.P.; (3) Total heat expended per hour per H.P.; (4) Steam efficiency; and (5) Heat rejected to condenser per lb. of steam used. *Ans.* (1) 69·6 B.T.U. (2) 37 lbs. nearly; (3) 40,381·8 B.T.U.; (4) 6·5 per cent.

9. What do you understand by "saturated steam" and "specific volume" of steam? A locomotive has two cylinders each of 18 inches diameter, the crank-arm measures 13 inches, and the engine makes 200 revolutions per minute. If the initial *gauge* pressure of the steam is 160 lbs. per square inch and it is cut off at $\frac{1}{4}$ of the stroke, how many gallons of water would be required per hour for the supply of the boiler? Neglect all losses from condensation and leakage. 1 lb. of steam at 175 lbs. per square inch measures 2·9 cubic feet. (S. & A., 1897, Adv.)

10. The calorific value of a fuel is 15·5 in standard evaporation units. How much steam at 300° F. will such a fuel produce if the feed water is at

60° F., and if all the heat could be utilised? Why is it impossible to utilise all the heat even in the most perfect boiler? (S. & A., 1898, Adv.)

11. What is the volume of 1 lb. of steam at 165° C., the latent heat being 490 in pound centigrade units? To find dp/dt approximately, use squared paper and the following information:—

θ° C., ...	160	165	170
Pressure in lbs. per square foot,	12,940	14,680	16,580

Prove your formula (B. of E., 1900, H., Part i.)

12. Describe a wire-drawing calorimeter for determining the wetness of the steam flowing along a steam pipe. What do you consider the chief difficulties in obtaining accurate results with such an appliance. In a test made with such an instrument the temperature of the wet steam was found to be 327·5° F., and after passing the wire-drawing orifice the temperature of the dried steam was 247·5° F. What was the wetness fraction for this steam? (C. & G., 1900, H., Sec. B.)

13. An engine uses 12·3 lbs. of steam per hour per H.P. developed. This steam is supplied to it superheated 150° F., and at a pressure of 150 lbs. absolute, the saturation temperature for such pressure being 358° F., the boiler feed temperature is 125° F.: calculate—(*a*) How many thermal units per hour per H.P. have to be supplied to the steam by the boiler and superheater (the total heat in a pound of saturated steam from a feed temperature of 32° F. is $= 1{,}082 + 0{\cdot}3\,t$ thermal units, t° being the temperature of the steam, and the specific heat of steam at constant pressure may be taken as 0·48); (*b*) How many thermal units are converted per hour per H.P. into work; (*c*) The thermal efficiency of the engine. (C. & G., 1900, Sec. C.)

14. A steam electric generator on three long trials, each with a different point of cut-off on steady load, is found to use the following amounts of steam per hour for the following amounts of power:—

Lbs. of steam per hour, . .	4,020	6,650	10,800
Indicated horse-power, . .	210	480	706
Kilowatts produced, . . .	114	290	435

Find the indicated horse-power and the weight of steam used per hour when 330 kilowatts are being produced. Find in the four cases the amounts of steam used per Board of Trade unit (that is, per kilowatt hour). (B. of E., 1901, Adv.)

15. An engine uses 4,000 lbs. of wet steam per hour at 170° C., there being 90 per cent. steam and 10 per cent. water. If the feed water was at 20°, how much heat is supplied? If the indicated horse-power is 140, how much heat energy is indicated per hour? If we imagine no heat to be radiated, and if the circulating water of the condenser is raised 10° C., how many lbs. of circulating water are being used per hour? (B. of E., 1901, Adv.)

16. In a central station it is found that during seven days' working

168 tons of coal, costing 21s., 3d. a ton, have been burnt, and that 190,400 gallons of feed water have been used, the price of the water being 4d. a thousand gallons. The energy developed and sold during this period is equivalent to 84,320 H.P.-hours. Find (*a*) the coal used per H.P.-hour; (*b*) the feed water evaporated per pound of coal; (*c*) the steam used per H.P.-hour; (*d*) the cost in fuel and water per H.P.-hour. (C. & G., 1901, O., Sec. C.)

17. Using the formulæ on the examination tables given you; find the heat given to 1 lb. of feed water at 40° C. to convert it into wet steam (15 per cent. water) at 170° C. If 25 lbs. of this wet steam reaches the cylinder per horse-power hour, what percentage of the heat leaves with the exhaust or is radiated from the cylinder? (B. of E., 1902, Adv.)

18. A given kind of coal, when burnt in one furnace, is found to evaporate 9 lbs. of water per lb. of coal from 60° F. at 324° F., and when burnt in a second furnace evaporates 8·5 lbs. of water from 104° F. at 350° F. The steam from the first boiler contains 10 per cent. of moisture and from the second 5 per cent. Compare the evaporative efficiencies of the two boilers. (Note, $H = 1082 + \cdot 305\, t$.) (C. & G., 1902, O., Sec. C.)

19. In the high-pressure cylinder of a compound engine it was found that the pressures (in lbs. per square inch absolute) at the points of cut-off, release, and compression were 64, 15·2, and 14·8 respectively, whilst the volumes as given by the card (including the clearance volume) were 2·92, 13·24, and 1·52 cubic feet respectively. The hot well discharge per minute was 29 lbs., and the number of working strokes 48. Assuming the steam shut in at the point of compression to be dry, find the dryness fractions of the steam at the points of cut-off and release, having given that the volumes of 1 lb. of dry steam at the pressures of cut off, release, and compression are 6·6, 25·5, and 26·3 cubic feet respectively. (C. & G., 1902, H., Sec. B.)

20. In order to test the dryness fraction of the steam supplied by a boiler, steam is led from the boiler into a tank containing a known weight of water, and the temperature before and after the steam is admitted, and also the quantity of steam admitted, are accurately determined In one trial the boiler pressure was 174 lbs. absolute (corresponding temperature 370° F.) and the initial and final temperatures of the water were 74° F. and 102·2° F. respectively. The weight of water originally in the tank was 4,085 lbs., and the weight of the steam blown in was 109½ lbs. The weight of the tank itself was 876 lbs., and the specific heat of the material of which it was composed was ⅛. Assuming that the water and the tank are at a uniform temperature, find the dryness fraction of the steam ($H = 1{,}082 + \cdot 305\, t$). (C. & G., 1902, H., Sec. B.)

21. In connection with the steam or gas or oil or spirit engine work with which you are acquainted, there is testing of some sort to be done requiring careful measurement of work or heat. For example, finding the calorific value of coal, gas, or oil; finding the latent heat of steam, or how its pressure depends upon temperature; finding the wetness of steam during an engine test; comparing the power of an engine and the quantity of heat or of steam or gas or oil used per hour. Describe, with sketches, some one such test. (B. of E., 1903, Adv. and H., Part i.)

22. When comparing different boilers, what do we take as the standard of evaporation? Feed water, 25° C.; steam, 15 per cent. wet—that is, there is 0·15 lb. of water to 0·85 lb. of steam leaving a boiler at 180° C. If 9 lbs. of this wet steam leaves a boiler for every lb. of coal burnt in the furnace, what is the evaporative value of the coal, reduced to standard units of evaporation? (B. of E., 1903, Adv.)

23. What is the cause of priming in boilers? Even if the boiler does not prime, why may wet steam reach the cylinder? What may be done to prevent it? Even if only dry steam enters the cylinder, why may there be condensation on admission? Why is this harmful? What may be done to prevent it? (B. of E., 1903, Adv.)

24. Feed water, 25° C.; steam, 10 per cent. wet—that is, there is 0·1 lb. of water to 0·9 lb. of steam at 170° C. If 25 lbs. of this wet steam enter the cylinder per hour per indicated horse-power, how much of the heat passes to the exhaust? If the stuff leaves the cylinder as saturated steam and water at 105° C., what is its wetness? Neglect radiation or other loss of heat by the cylinder. (B. of E., 1903, H., Part i.)

25. Given the following numbers for steam, use squared paper to find dp/dt at 150° C. The latent heat of steam at 150° C. is 500·8 in pound Centigrade units, find the volume of 1 lb. of steam at 150° C.

$\theta°$, ...	145	150	155
Pressure in lbs. per square foot,	8,698	9,966	11,380

Prove your formula. (B. of E., 1903, H., Part i.)

26. Describe, with sketches, any form of calorimeter with which you are acquainted for determining the dryness fraction of boiler steam. In a combined separating and wire-drawing calorimeter the following observations were made:—Quantity of steam obtained from separator, 2 lbs.; quantity of steam condensed after wire-drawing, 38 lbs.; steam pressure before wire-drawing, 102 lbs. abs.; corresponding temperature (from tables), 329° F.; steam pressure after wire-drawing, 15 lbs. abs.; corresponding temperature (from tables), 213° F.; actual temperature after wire-drawing, 240° F. Assuming the specific heat of superheated steam to be constant and equal to ·48, estimate the dryness fraction of the original steam. (C. & G., 1903, H., Sec. B.)

LECTURE XII.

CONTENTS.—Pressure and Volume of a Gas—Boyle's Law—Pressure, Volume, and Density—Watt's Diagram of Work, with Examples—Questions.

Pressure and Volume.—We saw in Lecture VII., by the experiment with Marcet's boiler and from Regnault's tables, that the pressure of steam increased with the temperature; we now come to consider the relation which exists between *pressure and volume.*

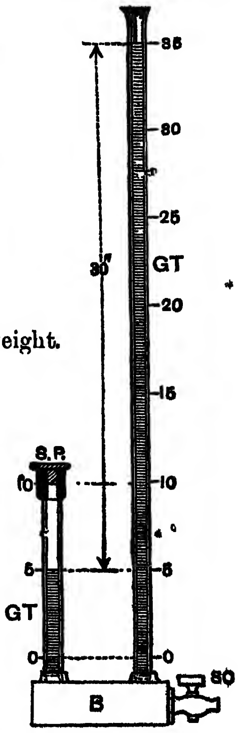
To understand this we here state the *first law* in regard to the expansion of gases, viz., Boyle's, and then give a class experiment to prove it.

Boyle's Law.—The pressure of a portion of a (perfect) gas *at a constant temperature* varies inversely as the space it occupies.

Or, let p = pressure in lbs. per sq. in.

v = volume in cub. ft. per lb. weight.

Then $p\,v$ = constant.



G T for Glass tubes.
B „ Box (air tight).
S C „ Stop-cock.
S P „ Screw plug.

To illustrate this law the following simple piece of apparatus may be used:—

It consists of a small metal box, B, to which are attached two glass tubes, G T, one a little more than 35″ long, and the other fully 10″. A stop-cock, S C, is screwed into the metal box, and the short tube is provided with a screw plug, S P. The whole is fixed to a board, on which there is a graduated scale of inches.

Mercury is poured into the long tube and the screw plug, S P, is taken out until the mercury rises in both tubes to the zero line. The screw plug is then replaced and encloses a column of air 10″ high in the short tube. Supposing the barometer to stand at 30″, we now continue pouring mercury into the long tube until the level of the mercury in it is 30″ above the

level of the mercury in the short tube. When this point, 35", is reached, the mercury in the short tube will be found to stand at 5". The air in the short tube has thus been subjected to an additional pressure of 30" of mercury, *i.e.*, to an additional pressure of one atmosphere; therefore, its pressure has been doubled. Before applying this pressure it occupied 10" of the tube; hence we see that its volume has been reduced by one-half by doubling the pressure on it, in accordance with the law just stated. It is important that the student should not overlook the fact, that this law is true, *only* when the temperature remains constant.

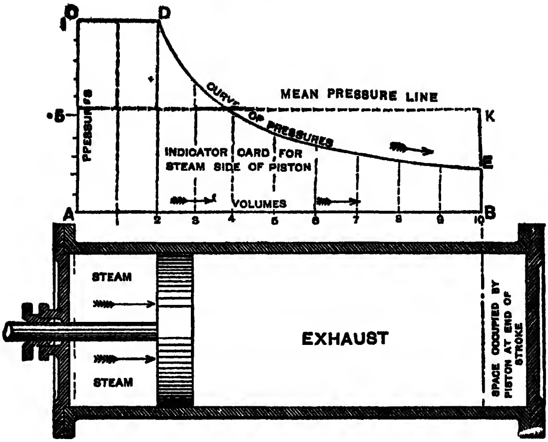
Since the pressure of an enclosed perfect gas kept at a constant temperature varies *inversely* as its volume, and since the density or weight per unit volume of the same, varies *inversely* as its volume, it follows that the pressure varies *directly* as the density.

This law is not perfectly fulfilled by any actual gas, but very nearly so by those gases which cannot be condensed into liquids, such as air. When a gas is about to pass by condensation into a liquid (*e.g.*, steam on the point of being transformed into water), then the density increases more rapidly than the pressure.

Watt, however, assumed that Boyle's law held good in the case of steam, and he applied it in a most ingenious manner to prove the economy of the expansive working of steam in a cylinder, and to show that he could get a greater amount of work from the steam by cutting it off early in the stroke, and thus allowing it to force the piston forward during the remainder of the stroke, merely by expansion.

Watt's Diagram of Work.—Although, as we shall see later on, steam does not expand in strict accordance with Boyle's law (for the temperature of the steam falls the more it is expanded, unless external heat is applied to it, to make up for the loss due to the work got out of it), yet we shall gain a great insight into the action of steam in an engine cylinder, by first discussing "Watt's Diagram of Work done during Expansion," and then applying the corrections that have since been found necessary, in order to truthfully represent the actual state of matters.

The following figure will illustrate to the student the method adopted by Watt. The horizontal line, or abscissa, A B, indicates the length of the stroke, and is divided into 10 equal parts; the vertical line, or ordinate, A C, represents the pressure of steam used by Watt, say one atmosphere, and is also divided into 10, or decimal parts of an atmosphere of pressure. When the piston has travelled the distance, C D, *i.e.*, $\frac{2}{10}$ or $\frac{1}{5}$ of the stroke, the steam is cut off, and the remainder of the stroke is effected by



WATT'S DIAGRAM OF WORK.

the expansive action of the steam. The gradually falling curve, D E, marked "curve of pressures," is found by drawing verticals from each of the divisions of the stroke, 3, 4, ... 10, and marking them off in height corresponding to the pressures, p, at these points by the following formula, and joining their upper ends by a curved line:—

$$p\,v = \text{a constant, or } p = \frac{\text{constant}}{v}.$$

Where v = the volume swept out by the piston at the several points, and is, therefore, represented by the different distances, 2, 3, ... 10, from the commencement of the stroke.

For example—

At point of cut off
$p = 1$
$v = 2$
$\therefore$ Constant $= p\,v$
" $= 1 \times 2$
" $= 2$

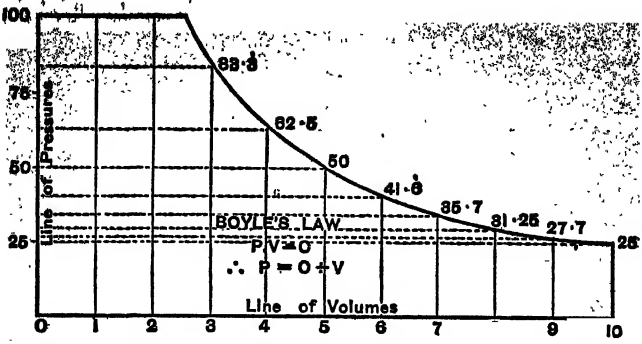
						Atmosphere.
At point 1, p		. . . . .				1
" 2, p		. . . . .				1
" 3, p	=	$\frac{\text{constant}}{v}$	=	$\frac{2}{3}$	=	0·66
" 4, p	=	"	=	$\frac{2}{4}$	=	0·5
" 5, p	=	"	=	$\frac{2}{5}$	=	0·4
" 6, p	=	"	=	$\frac{2}{6}$	=	0·33
" 7, p	=	"	=	$\frac{2}{7}$	=	0·29
" 8, p	=	"	=	$\frac{2}{8}$	=	0·25
" 9, p	=	"	=	$\frac{2}{9}$	=	0·22
End of stroke, " 10, p	=	"	=	$\frac{2}{10}$	=	0·2
Dividing by the Number of Parts, viz.,					10	4·85
We get roughly a Mean Pressure					=	·485*

*This mean pressure is less than the true mean as explained at next page and in Lecture XVII.

By adding the several pressures, and dividing them by the number of divisions taken—viz., 10—we get the average pressure throughout the stroke, = ·485 of an atmosphere, or nearly half an atmosphere. The economy of cutting off the steam before the end of the stroke will, therefore, be at once apparent, for we have obtained an average pressure equal to nearly half that which would have been obtained by carrying full steam pressure throughout the whole stroke, and have only used $\frac{1}{5}$ of the quantity of steam.

Since work done is measured by force or pressure, multiplied by the distance through which the force or pressure acts, the area of the rectangle, A D (see upper part of previous Fig.), being equal to the pressure, A C, if reckoned in lbs., multiplied by the distance, A 2, or, C D in feet, measures to scale the work done upon the piston by the steam up to the point of cut-off in foot-pounds or units of work. In the same way, the area of the rest of the figure—viz., D E B 2, measures to scale the work done upon the piston by the steam while expanding in the cylinder, also in foot-pounds; for this area is equal to the mean pressure in lbs. between the points, D and E, multiplied by the distance, 2 B, in feet. Consequently, the area of the *whole* figure, A C D E B, measures to scale the *whole* work done by the steam in one stroke in foot-pounds. This area is equal to the calculated mean pressure throughout the stroke, multiplied by the whole stroke, A B, and expresses the result of Watt's diagram of work. Watt, in calculating the mean pressure throughout the stroke, assumed that the pressure at each of the points into which he divided the stroke commencing with number 1, remained constant until it arrived at the next in order, by which method he obtained a less value than the true mean, because he omitted to take into account the ordinate of pressure at the point, A, or the very commencement of the stroke. If we now take into account the first ordinate at A, as well as the last one at 10, we have the following eleven pressures:—1, 1, 1, $\dot{6}$, ·5, ·4, $\dot{3}$, ·29, ·25, $\dot{2}$, and ·2, giving a total sum of 5·86, which sum being divided by the number of ordinates, viz., 11, gives us a mean of ·532 of an atmosphere instead of ·485, or nearly 8 lbs. pressure on the square inch, which is a nearer approximation to the true mean.

Let us take another example of Watt's diagram of work, taking the first as well as the last pressure ordinate into account, in order to get a nearer approximation to the true mean. Suppose we have an engine using steam of 100 lbs. pressure per square inch, and cutting off at $\frac{1}{4}$ of the stroke, to find the curve of expansion by Boyle's Law and the mean pressure.



As before—

	At 0, p	. . . . . .			100	lbs.
	At point 1, p	. . . . . .			100	„
	„ 2, p	. . . . . .			100	„
	„ 3, $p =$	$\frac{\text{constant}}{v}$	$= \frac{25}{\cdot 3}$	=	$83\cdot\dot{3}$	„
	„ 4, $p =$	„	$= \frac{25}{\cdot 4}$	=	62·5	„
Constant $= p\,v$	„ 5, $p =$	„	$= \frac{25}{\cdot 5}$	=	50	„
„ $= 100 \times \frac{1}{4}$	„ 6, $p =$	„	$= \frac{25}{\cdot 6}$	=	$41\cdot\dot{6}$	„
„ $= 25$	„ 7, $p =$	„	$= \frac{25}{\cdot 7}$	=	35·7	„
	„ 8, $p =$	„	$= \frac{25}{\cdot 8}$	=	31·25	„
	„ 9, $p =$	„	$= \frac{25}{\cdot 9}$	=	$27\cdot\dot{7}$	„
	„ 10, $p =$	„	$= \frac{25}{1}$	=	25	„

Dividing by the Number of Points, viz., 11 | 657·2

We get an approximate Mean Pressure = 59·7 lbs.

There are several rules for obtaining approximately the mean pressure from a diagram of work such as we have been discussing. The plan most commonly adopted by engineers (as we shall see at Lecture XVI.) in finding the mean pressure from actual indicator diagrams is, to measure by a suitable scale or rule the length of each of the ten ordinates, taken at the centre of each of the ten spaces into which the diagram is divided, add them together, and divide by their number. For instance, applying this rule to the last example, we should measure the length of the vertical lines midway between the points 0 and 1, 1 and 2, 2 and 3, ... 9 and 10, add these ten pressure ordinates

together, and divide the sum by 10, to get the mean pressure; and doing so (or calculating these pressures by pv = constant), we find them to be respectively, 100, 100, 100, 71·43, 55·5, 45·45, 38·46, 33·3, 29·41, and 26·31 lbs., giving a mean of 59·9 lbs., or slightly greater than that found above.

Simpson's Rule is as follows:—Divide the length of the figure into n equal parts, n being an even number, and draw ordinates through the points of division to touch the boundary lines. Add together the first and the last ordinates, call the sum A; add together the even ordinates 2, 4, 6, &c., call the sum B; add together the odd ordinates 3, 5, 7, &c., except the first and the last, and call the sum C; then $\frac{A+4B+2C}{3n}$ = mean ordinate or pressure. This quantity multiplied by the length, L, of the figure gives the area of the figure, or what we would call the area of work in this case.

Methods of Constructing the Curve of Pressures and Volumes by Boyle's Law.—We shall now show how to construct the curve for the relation between pressure and volume of a perfect gas expanding according to Boyle's law. This curve may be constructed in two different ways:—

1. By making use of the formula expressing Boyle's law—viz., pv = a constant, and thus calculating the pressure at various points during the expansion.

2. Or, we may adopt a purely graphical method for determining a series of points on the curve. The *curve of expansion* can then be drawn freehand or by aid of French curves, or by bending a thin flexible strip of wood until its lower edge passes through the several points. These two methods will be clearly understood from the solution of the following example.

EXAMPLE I.—Steam is admitted into the cylinder of an engine at a pressure of 30 lbs. by gauge, and is cut off at $\frac{1}{8}$ of the stroke. Draw to scale the diagram of work done during admission and expansion, assuming that the steam expands according to Boyle's law. From the diagram thus constructed, find the pressures at $\frac{3}{8}$, $\frac{5}{8}$, and $\frac{7}{8}$ of the stroke respectively.

ANSWER.—**First Method, by Calculation.**

Draw two axes O P, O V, at right angles to each other. Along O P, measure off a distance O A, to represent the initial absolute pressure of the steam.*

* The initial pressure as given by the question is 30 lbs. by gauge. The pressure as indicated by a steam gauge on a boiler or cylinder of an engine, has for its starting (or zero) point, the pressure of the atmosphere—viz., about 15 lbs. per square inch. We cannot, therefore, base our calculations respecting a law of nature on such an arbitrary and variable starting-point as this. Consequently, we must refer all our pressures to the *absolute zero* or perfect vacuum line before applying Boyle's law. The absolute zero is

Along O V, measure off a distance O B, to represent the volume of the stroke.*

Divide O B into any number of parts, equal or unequal in length, and at each point of division raise a perpendicular line of indefinite length. In the figure we have divided the

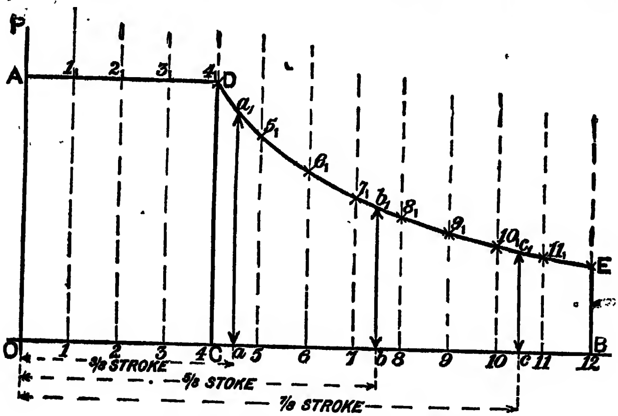


DIAGRAM OF WORK FOR EXAMPLE I.

stroke into 12 *equal* parts, for the following reasons: (1) the number 12 is a multiple of the denominator of the fraction $\frac{1}{3}$ (the fraction of the stroke at cut-off), so that one of the points of division may coincide with the point representing the cut-off; (2) the number of points on the curve will thus be sufficiently numerous or close together to enable us to draw a fairly accurate curve through them; and (3) we have taken the parts of equal length because the stroke will be divided into convenient and easily recognised fractions. We might have divided the stroke into *nine* or even *ten* equal parts, as is usually the case in practice, but, due to the reasons just assigned, we prefer the larger number for the present case.

Denote the pressures and volumes at the points 0, 1, 2, 3 . . . . 12, by the letters p_0, v_0, p_1, v_1, p_2, v_2, p_3, v_3, ... p_{12}, v_{12} respectively.

thus about 15 lbs. (more correctly 14·7 lbs.) per square inch below atmospheric pressure. Hence, the initial absolute pressure of steam = 30 + 15 = 45 lbs. per square inch. We have, therefore, to make O A represent this total pressure.

* Since the cross area of the cylinder is constant throughout the stroke, the line O B will also represent to scale the full stroke of the piston, and the distances O_1, O_2, O_3, &c., definite proportions of the stroke.

Further, let the whole volume of the piston's stroke be denoted by the number 12—*i.e.*, let $v_{12} = 12$. Then, $v_1 = 1$, $v_2 = 2$, and so on. The utility of this notation will be apparent from the following.

The point of cut off coincides with the point 4 ($\frac{1}{3} \times 12 = 4$), as shown by the figure. Now we know that

$$p_4 = p_0 = 45 \text{ lbs. absolute; and that } v_4 = 4,$$
$$\therefore \text{ by } \textit{Boyle's Law}, \; pv = \text{a constant}$$
$$\therefore \quad \text{The Constant} = p_4 v_4 = 45 \times 4 = 180.$$

Calculate and tabulate the pressures at the various points during expansion, thus—

$$p_5 = \frac{const.}{v_5} = \frac{180}{5} = 36\cdot00 \text{ lbs. abs.}$$
$$p_6 = \frac{const.}{v_6} = \frac{180}{6} = 30\ 00 \quad ,, \quad ,,$$
$$p_7 = \frac{const.}{v_7} = \frac{180}{7} = 25\cdot71 \quad ,, \quad ,,$$
$$p_8 = \frac{const.}{v_8} = \frac{180}{8} = 22\ 50 \quad ,, \quad ,,$$
$$p_9 = \frac{const.}{v_9} = \frac{180}{9} = 20\ 00 \quad ,, \quad ,,$$
$$p_{10} = \frac{const.}{v_{10}} = \frac{180}{10} = 18\ 00 \quad ,, \quad ,,$$
$$p_{11} = \frac{const.}{v_{11}} = \frac{180}{11} = 16\ 36 \quad ,, \quad ,,$$
$$p_{12} = \frac{const.}{v_{12}} = \frac{180}{12} = 15\ 00 \quad ,, \quad ,,$$

We now possess all the data for completing the diagram. Along the perpendiculars drawn through the points 4, 5, 6 . . . . 12, measure off distances 4, 4_1, 5, 5_1, 6, 6_1 . . . . 12, 12_1 respectively, to represent (according to the scale previously employed for the pressure O A) the pressures p_4, p_5 . . . . p_{12}, given above. Then 4_1, 5_1 6_1 . . . . 12_1, are points on the expansion curve. Join A with point 4_1, and through the points 4_1, 5_1, 6_1 . . . . 12_1, draw carefully by hand (or otherwise as previously directed) an unbroken continuous curve, D E. This is the expansion curve, and is known to mathematicians as a *Rectangular Hyperbola*. The area of the rectangle O A D C represents the work done to the point of cut off; the area of the figure C D E B represents the work done during expansion. The area

of the whole figure O A D E B represents to scale the complete diagram of work.

We are also asked by the question to find, from the diagram thus constructed, the pressures at $\frac{3}{8}$, $\frac{5}{8}$ and $\frac{7}{8}$ of the stroke.

Since the length of stroke has been denoted by the number 12, we, therefore, get—

$$\tfrac{3}{8} \text{ stroke} = \tfrac{3}{8} \times 12 = 4{\cdot}5$$
$$\tfrac{5}{8} \quad " \quad = \tfrac{5}{8} \times 12 = 7{\cdot}5$$
$$\tfrac{7}{8} \quad " \quad = \tfrac{7}{8} \times 12 = 10{\cdot}5$$

These points are easily found, and are indicated on the right-hand part of the figure by the letters *a*, *b*, and *c* respectively. Drawing the *ordinates*, aa_1, bb_1, cc_1, and measuring their lengths, we get, according to the scale of pressures, the following results—

$$aa_1 = 40{\cdot}00 \text{ lbs. abs.}$$
$$bb_1 = 25{\cdot}00 \quad "$$
$$cc_1 = 17{\cdot}14 \quad "\ *$$

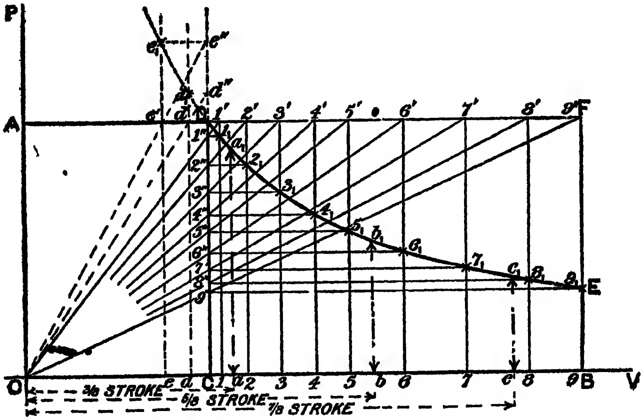
Second Method: by Graphical Construction.—As before draw two axes O P, O V, and measure off the distances O A, O B to represent the initial pressure and the volume of stroke respectively. Let O C represent the volume swept through by the piston to the point of cut off. Complete the rectangles O A D C, O A F B. Divide C B into any number of equal or unequal parts, and at these several points of division raise perpendiculars to meet the line A F.†

To find points on the expansion curve, join the origin O, with the point 1′, on the line A F. This line cuts the perpendicular C D, in the point 1″. Through point 1″, draw the line 1″ 1_1, parallel to O V, and terminated by the perpendicular through point 1. *The point* 1_1, *is a point on the expansion curve.* Similarly, join O 2′. This line O 2′, cuts the perpendicular C D in the point 2″. Through 2″, draw the line 2″2_1, parallel to O V, to meet the perpendicular 2, 2′, in point 2_1. *Then, point* 2_1, *is also a point on the expansion curve.* By proceeding in this way, as shown by the figure, we get the series of points 1_1, 2_1, 3_1 . . . .

* These are the exact values, as may be readily proved by calculation. When, however, the measurements are carefully made, and the curve neatly drawn, such results should not differ from the correct results by more than 2 or 3 per cent.

† An inspection of the curve D E (in the previous figure) shows that it is steeper near to the end D than it is towards the end E. Hence, if we use equidistant ordinates, a greater length of curve will lie between two consecutive points near the end D than towards the flatter portion of the curve at E. For this reason, it is advisable to have the points 1, 2, 3 . . . near to C, much closer together than those points towards the end E.

on the curve. The curve can then be drawn through the points thus found.



GRAPHIC CONSTRUCTION FOR FINDING POINTS ON EXPANSION CURVE.

The pressures at $\frac{3}{8}$, $\frac{5}{8}$, and $\frac{7}{8}$ of the stroke can then be found as before, by measuring the ordinates through the points *a*, *b*, and *c* respectively.

If the steam was *compressed*, according to Boyle's law, from an initial volume O C, the curve of compression would be a continuation of the curve E D, as shown in dotted line by the figure. The method of drawing the compression curve is identically the same as that described above for the expansion curve. It will be sufficient to show how to find one point on this compression curve.

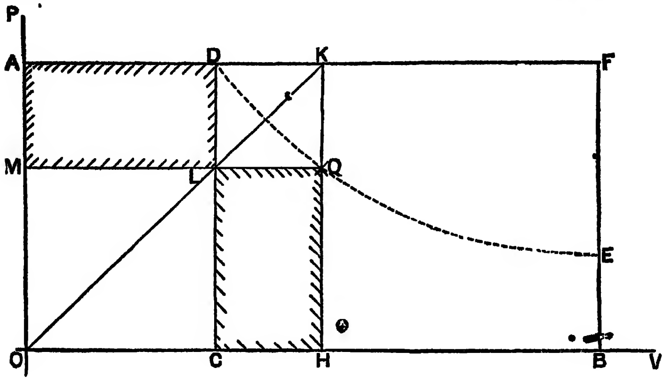
Suppose we require to find the pressure when the volume is diminished to O *d*. Through *d*, draw the perpendicular $D\,D_1$, cutting A F, in the point d'. Join O d', and produce it to meet C D, produced in d''. Through d'', draw $d''d_1$, to meet $d\,d_1$, in d_1 *Then* d_1 *is a point on the compression curve.*

Proof of above Construction.—That the above construction is mathematically correct may be proved as follows:

Let O A = Initial pressure, p_1,
O B = Volume of stroke,
O C = Volume to cut off, v_1.

Let the pressure and corresponding volume at any other point H

of the stroke be represented by the letters p_h and v_h respectively. Complete the rectangles O A D C, O A F B. Find by the previous construction the point Q, on the expansion curve corresponding



ILLUSTRATING PROOF FOR GEOMETRICAL CONSTRUCTION.

to volume O H. Produce Q L to meet O P in M. Then the area of the rectangle O A D C represents the product, $p_1\ v_1\ h$, *i.e.* the pressure × the volume at the point C. We have now to prove that—*area of rectangle* O M Q H = *area of rectangle* O A D C.

Since O A K H, is a parallelogram having parallelograms, O M L C, L D K Q, described on its diagonal O K, then the remaining parallelograms M A D L, and C L Q H (which make up the complete figure O A K H) are called *complementary parallelograms*. Now, by *Euclid, Book I. proposition* 43, it is proved that the areas of complementary parallelograms are equal. Hence the parallelogram C L Q H = the parallelogram M A D L. Add to each side of this equation the parallelogram O M L C, and we get—

$$\textit{Area } O\ M\ Q\ H = \textit{Area } O\ A\ D\ C$$
$$\textit{i.e. } \mathrm{O\ M \times O\ H = O\ A \times O\ C}$$
$$\text{Or,} \quad p_h \times v_h = p_c \times v_c$$

Which proves that Q is a point on the expansion curve.

Simpler Proof.—The following is a still simpler proof. In the similar triangles O L C, and O K H, we get—

$$\mathrm{C\ L : C\ O = H\ K : H\ O} \quad \text{(Euclid VI.–2.)}$$
$$\therefore \mathrm{C\ L \times H\ O = H\ K \times C\ O}$$
$$\text{But, } \mathrm{C\ L = H\ Q, \text{ and } H\ K = O\ A}$$
$$\therefore \mathrm{H\ Q \times H\ O = O\ A \times O\ C}$$
$$\therefore \text{As before } p_h \times v_h = p_c \times v_c$$

Lecture XII.—Questions.

1. State Boyle's law, and describe an experiment to show that the pressure of a gas varies inversely as the space it occupies.

2. Steam is admitted into a cylinder at atmospheric pressure, and is cut off at half stroke. Divide the stroke into 10 equal parts, and, supposing that the pressure at the beginning of each of these portions remains uniform until the piston reaches the next in order, find the pressure at each point as well as the mean pressure.

3. The cylinder of an engine is 25 inches long, and steam is admitted at 18 lbs. total pressure, the final pressure being 4 lbs. At what point of the stroke was the steam cut off? *Ans.* 5·5 inches.

4. Steam is admitted into a cylinder at a pressure of 25 lbs. on the square inch above the atmospheric pressure of 15 lbs. on the square inch, and is cut off at such a point that its pressure at the end of the stroke is 5 lbs. below that of the atmosphere. At what point of stroke was it cut off? Make a diagram, showing approximately the steam pressure on the piston throughout the stroke. *Ans.* ·25 of the stroke.

5. The cylinder of an engine is 25 inches long; steam is admitted at 18 lbs. actual pressure, and the final pressure is 4 lbs. Divide the stroke into 10 equal parts; find the steam pressure at each point of division, and set out Watt's diagram of work done. Find also the mean pressure of the steam by Watt's, by Simpson's, and by the usual rule. *Ans.* mean = 10 lbs.

6. The stroke of a piston is 4 feet 6 inches, the steam is cut off at 9 inches, and the pressure at the end of the stroke is 5 lbs. below that of the atmosphere. At what pressure above the atmosphere was steam let in? 45 lbs.

7. Steam is admitted into the cylinder of an engine at a pressure of 45 lbs. per square inch by gauge, and is cut off at one-third of the stroke. Find the pressure in pounds at half-stroke, and also at the end of the stroke. Show roughly, by a diagram, that additional work is obtained from a given quantity of steam—(1) by cutting off the supply from the boiler before the end of the stroke; (2) by condensing the steam instead of allowing it to escape into the air. *Ans.* 25 lbs.; 5 lbs. above atmosphere.

8. Explain the advantage of working steam expansively and with condensation. Steam is admitted into a cylinder at 30 lbs. above the atmosphere, which is taken at 15 lbs. per square inch, and is cut off at a certain point, and then expands to a pressure of 5 lbs. below the atmosphere. If the length of stroke be 4½ feet, at what point is the steam cut off? *Ans.* 1 foot.

9. The temperature of a condenser is 100° F., and the corresponding pressure from Regnault's tables is ·942 lbs. The vacuum shows 26 inches by gauge, and the barometer stands at 29 9 inches; what part of the increase in back pressure is due to air in condenser? *Ans.* = ·03 lb.

10. The mean steam pressure on a piston being 26 lbs. to the square inch above atmospheric pressure, and the mean vacuum pressure 13·5 lbs. to the square inch, what is the total force exerted on a piston 63 inches in diameter? What would have been the force if the engine had exhausted at atmospheric pressure? *Ans.* 123,131 lbs.; 81,048 lbs

LECTURE XIII.

CONTENTS.—Charles' Law of the Expansion of Gases—Absolute Zero of Temperature—Expansion of a Gas doing External Work—Adiabatic Expansion—Heat Engines—Carnot's Principle—Entropy and Thermodynamics from an Engineer's Point of View—Questions.

Second Law of the Expansion of Gases.—This law, which was discovered by Charles, and is known as Charles' law, was first published by Dalton in 1801, and independently by Gay Lussac in 1802. It may be stated as follows:—

A gas, under constant pressure, expands by a definite fraction of its volume at 32° Fah., for a given increase of temperature. The amount of this increase of volume has been the subject of careful investigation by many experimenters, and the value assigned by Regnault is that the expansion of a gas between 32° Fah. and 212° Fah. is ·3665 of its volume at 32° Fah. We must also note the remarkable fact that the amount of expansion is the same for all gases. The laws of the expansion of gases are only approximately true for actual gases, but form the essential characteristics of a *perfect gas* The variation from this second law is very slight in permanent gases (*i.e.*, gases which cannot be liquefied by cold or pressure), but is more considerable in liquefiable gases. Every gas, however, more nearly fulfils this law the more highly it is heated and rarefied. Air, when perfectly dry, deviates but slightly in its behaviour from that of a perfect gas, but when containing moisture, as it almost always does in practice, the deviation is considerable.

Absolute Temperature.—We have seen that a volume of gas falling in temperature from 212° Fah. to 32° Fah., contracts in the ratio of 1·3665 to 1. A curious question now arises, viz., At what temperature will the volume of the gas diminish to nothing? The gas in falling the 180° between boiling and freezing points on the Fahrenheit scale, decreases in volume ·3665 of its volume at freezing point, with what fall in temperature from 32° would it decrease to nothing?

Stating by proportion we have—

$$\cdot 3665 : 1 :: 180 : x$$

$$\therefore x = \frac{180}{\cdot 3665} = 491^\circ\cdot 13 \text{ Fahrenheit.}$$

Or, let x = temperature from freezing point;

Then—

$$\frac{x}{x+180} = \frac{1}{1\cdot 3665}$$

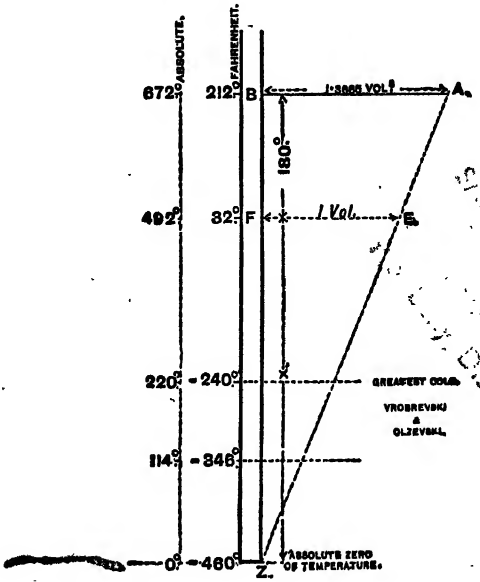
$$\therefore 1\cdot 3665\, x = x + 180$$

$$\cdot 3665\, x = 180$$

$$\therefore x = \frac{180}{\cdot 3665} = 491^\circ\cdot 13 \text{ F.}$$

That is, when the temperature has fallen 491°·13 Fah. *below* freezing point, or 459°·13 below the zero of Fahrenheit's scale, the gas will occupy no space at all, and all the heat will have been extracted from it. This number, − 459°·13, or practically − 460° Fah., is termed the *absolute zero of temperature*, and corresponds approximately to, − 273° Cent.

The annexed diagram* will make this clear.



BZ is a line of temperatures, F, being the freezing point and, B, the boiling point. We know that the ratio of the volume of the gas at F to the volume at B is $\frac{1}{1\cdot3665}$; therefore, if we draw at right angles to, BZ, two lines, BA, and, FE, to represent the relative volumes of the gas at these points, then join AE, and produce it beyond E, we find that it cuts the line of temperatures, BZ, at a point, Z, 492° below freezing point, showing that at that point the volume of the gas has been reduced to nothing.

* From *The Howard Lecture*—"On the Conversion of Heat into Useful Work," by William Anderson, M. Inst. C.E.

This absolute zero of temperature has been fixed solely by reasoning, and no temperature so low having ever been obtained, we can assert nothing as to the state of a gas when deprived of all its heat. In questions on thermodynamics and the expansion of gases, it is most convenient to measure temperatures, not from the arbitrary zero of the Fahrenheit scale, but from the absolute zero just found, so that, if t = temperature from Fahrenheit zero, τ = absolute temperature Fah. Then $\tau = t + 460$.

We now have a means of estimating the amount of expansion of a gas due to a given rise of temperature, the expansion being proportional to the absolute temperature. All gases expand $\frac{1}{492}$ of their volume at 32° Fah. for every increase of temperature of 1° Fah.* We may, therefore, express the laws of Boyle and Charles by the formula—*The product of the pressure and volume of any gas is proportional to the absolute temperature, i.e.*—

$$PV = 144pV = c\tau \quad \text{Or, } c = \frac{PV}{\tau}.$$

Where P = lbs. per sq. ft; V = vol in cub. ft; c = constant for the gas †

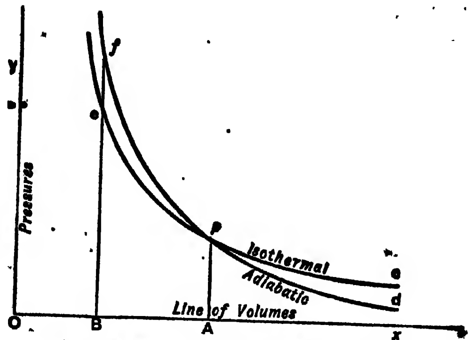
Expansion of a Gas doing External Work.—Hitherto we have been dealing with the expansion of a gas in accordance with Boyle's law, and we have drawn the curve of expansion in Watt's diagram of work. The necessary condition for the fulfilment of Boyle's law is, that the temperature remains constant, and the curve representing expansion under this condition is an hyperbola, and is known in connection with this subject as an "*isothermal curve," or curve of equal temperature.* Suppose we have an ideal engine, the cylinder of which is constructed of non-conducting material, so that no heat can enter or leave the working gas during its action, and that we introduce a mass of air at a certain pressure, volume, and temperature, and allow it to do work on the piston by its expansive power; although no heat can pass through the cylinder, yet we find that the temperature of the working gas falls considerably throughout the expansion. The explanation is easy. The gas in expanding converts a quantity of its heat into actual mechanical work, and the amount of work done by our ideal engine *must* be the exact mechanical equivalent of the heat lost by the gas in the cylinder. Therefore, when a gas expands doing external work, its temperature must fall (otherwise no work could be done), and the relation between pressure and volume will not be in accordance with Boyle's law, unless heat is supplied to the substance during expansion, in proportion to

* *i.e.*, The coefficient of expansion per degree for a constant pressure is, = the reciprocal of the absolute temperature of melting ice = $\frac{1}{492}$ on Fah. scale, or $\frac{1}{273}$ on Cent. scale.

† See Appendix E for values of this constant c for air and steam, and how these are found.

the amount of work done by the gas. This is a most important point, and is one of the claims for a steam jacket, such as that used by Watt when working steam expansively; we shall, however, have occasion to refer to this application of the principle later on. If a gas expands without doing any external work its temperature is unaltered.

Adiabatic Expansion.—*Expansion doing work without gain or loss of heat from an external source* (*as already referred to*) *is termed "adiabatic" expansion*, and the curve which represents the changes of pressure and volume throughout is termed an "adiabatic" curve, to distinguish it from the isothermal curve of expansion according to Boyle's law.

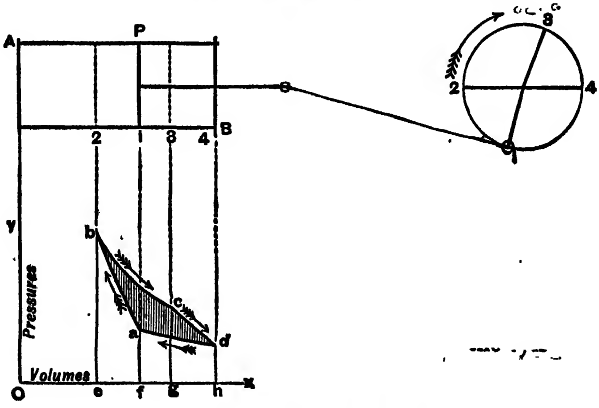


ISOTHERMAL AND ADIABATIC CURVES.

Suppose we have a volume, O A, of a gas at a pressure, A *p*, and expand it at a uniform temperature, and let the changes of pressure and volume be represented by the isothermal curve, *p c*. Then, if the gas be expanded adiabatically, the changes of pressure and volume will be represented by the adiabatic curve, *p d*, and it is evident that this curve must fall below the isothermal since the gas has done work. If, however, we compress the gas from the volume O A to the volume O B, we are now doing work upon it, and the equivalent of the work done upon the gas is imparted to it in the form of an additional quantity of heat. If the compression takes place at uniform temperature this additional heat must escape, and the changes of pressure and volume will be represented by the isothermal curve, *p e*. If, however, the compression takes place under the condition that no heat can escape, then the changes of pressure and volume will be represented by the adiabatic curve, *p f*, and this curve rises above the isothermal. For, the effect is the

same as if the gas had first been compressed at constant temperature and then a certain quantity of heat had been taken up by it while its volume was kept constant. Thus, we see that the adiabatic curves are more inclined to the line of volumes than the isothermals, and, therefore, to diminish the volume of a gas by a given amount, requires a greater increase of pressure when the gas is prevented from losing heat, than when it is kept at a constant temperature. We infer from this, that even in a conducting cylinder, if the compression takes place very suddenly, before the heat has time to escape, a greater pressure will be required to compress it than if the action were gradual. For a complete study of this part of the subject we would refer the student to the larger treatises.*

Heat Engines.—We propose now to discuss an ideal or imaginary engine, for the conversion of heat into mechanical work, and shall calculate the work done by the engine.



IMAGINARY HEAT ENGINE.

Let AB be the working cylinder of the heat engine, having a volume of a perfect gas, say air between the cover, A, and the piston, P. The cylinder must be constructed of non-conducting material, so as to allow of adiabatic expansion, but yet it must be capable of transmitting heat to and from the working substance at certain intervals of the stroke. We see, therefore, that these special conditions of working are contradictory, but,

* Such as Rankine on *The Steam Engine*, Cotterill on *The Steam Engine considered as a Heat Engine*, Maxwell on *Heat*.

as the engine is purely imaginary, we shall suppose that this difficulty has been overcome.

Starting then with the piston in position, 1, we have a volume of gas, O f (see diagram below cylinder), at a pressure, fa, and absolute temperature, τ_1. As the crank moves from position 1 to position 2 (upper diagram), the air is compressed, and, since no heat can escape, the relation* between pressure and volume is represented by the adiabatic curve, $a\,b$ (lower fig.) When the piston has reached point 2, the absolute temperature of the working substance has risen to τ_2. During the movement of the piston from point 1, to point 2, work has been spent upon the working substance, and this we may call *negative* work. The working substance now forces the crank further round and begins to expand. The effect of this would be to make the temperature fall, but throughout the expansion from 2 to 3, heat is supplied to the working substance in sufficient quantity to maintain the temperature uniform, and the relation between volume and pressure is represented by the isothermal curve, $b\,c$. During this process the substance is doing work, and this we reckon as *positive*. In order to maintain this isothermal curve a quantity of heat, H, has been taken up by the working substance. When the piston arrives at point 3, the supply of heat to the working substance is stopped, and the expansion continues without gain of heat, therefore the temperature falls. The relation between the pressure and volume at this part of the stroke, is represented by the adiabatic curve, $c\,d$. When the temperature has fallen to τ_1 (the temperature from which we started) the piston will have arrived at the end of its stroke, and the work having been done *by* the substance will be reckoned *positive*. The crank is now in position 4, and on passing that point causes the piston to move back and to compress the air in the cylinder. This compression would cause a rise of temperature, but the additional heat imparted to the substance is abstracted during the compression, and the relation between pressure and volume is exhibited by the isothermal curve, $d\,a$. Since, before beginning the return stroke the working substance had the same temperature as that from which we started, therefore, when the piston arrives at point 1, the working substance has returned exactly to its original state as regards volume, pressure, and temperature. During this latter portion of the stroke work has been spent upon the substance, and must, therefore, be considered *negative*, and simultaneously a quantity of heat (say h) has been abstracted from the working substance.

Such a series of operations as this, by which the working

substance, after undergoing successive states of pressure, volume, and temperature, is finally brought back in all respects to its original state, is termed a *cycle* of operations. When the changes of state can be passed through in *either* direction, the cycle is said to be *reversible*. In the diagram of the expansion of the gas, the figure, *a b c d*, represents the cycle, for, while we started with a pressure, *af*, volume, O*f*, and temperature, τ_1, we arrived at the close of our operations with the same pressure, volume, and temperature. The working substance is performing work on the piston while it moves from point 2, to point 4, the expansion curve being *b c d*, and the work done is represented by the area of the figure, *b c d h e*. This is positive work. Work is done upon the substance while the piston moves from point 4, to point 2, the expansion curve being *d a b*, and the work done is represented by the area of the figure, *d a b e h*. This is negative work. To find the work performed by the substance, we subtract the area, *d a b e h*, representing negative work, from the area, *b c d h e*, representing positive work, and the remainder, *a b c d*, represents the useful work performed by the substance throughout the cycle of operations.

Work done by a Heat Engine.—Our operations on this heat engine consisted in taking in a quantity of heat, H (from positions 2 to 3 of stroke), at a temperature, τ_2, and rejecting a less quantity of heat, *h* (from 4 to 1), at a lower temperature, τ_1.

Hence, the heat-energy that has been transformed into mechanical work during this cycle of operations

$$= \mathrm{H} - h \text{ thermal units.}$$

Calling W the amount of work thus given out by the engine in one revolution, we have

$$\mathrm{W} = \mathrm{J}\,(\mathrm{H} - h) \text{ foot-lbs.,}$$

$$= \mathrm{J\,H}\left(1 - \frac{h}{\mathrm{H}}\right) \quad ,,$$

where J is Joule's "equivalent" (see Index).

Now, in order to eliminate *h* from the above equation, and thus express the work actually done in terms of the heat supplied, together with the absolute temperatures between which the engine works, we proceed as follows:—

$$\begin{array}{llll} \text{Let } p_1 = & \text{pressure measured by} & & fa, \\ p_2 = & ,, & ,, & eb, \\ p_3 = & ,, & ,, & gc, \\ p_4 = & ,, & ,, & hd, \end{array}$$

v_1, v_2, v_3, and v_4 be the corresponding volumes occupied by the gas when subjected to the above pressures respectively.

In Lecture XVII. it is shown that when a gas expands, according to Boyle's law, from a pressure and volume, p_2, v_2, to another pressure and volume, p_3, v_3, say, and does work, the value of the work done is

$$= p_2 v_2 \log_e \frac{v_3}{v_2} \text{ foot-lbs.},$$

when p_2 is in lbs. per sq. ft., and v_2 is in c. ft.

Now, since our working substance, the gas, takes in a quantity of heat, H, while expanding from a volume, v_2, to a volume, v_3, sufficient in amount to keep the temperature at the constant value, τ_2, it follows, from what has been already said about isothermal expansion, that the work done on external bodies by the gas, must be equal to the mechanical equivalent of the heat supplied, that is to say,

$$\text{J H} = p_2 v_2 \log_e \frac{v_3}{v_2},$$

$$= c \tau_2 \log \frac{v_3}{v_2}.$$

By similar reasoning, we get

$$\text{J } h = c \tau_1 \log_e \frac{v_4}{v_1}.$$

Substituting the values of H and h, thus obtained into the equation for the work done, we get

$$\text{W} = \text{J H} \left\{ 1 - \frac{\tau_1 \log \frac{v_4}{v_1}}{\tau_2 \log \frac{v_3}{v_2}} \right\}.$$

Since the points, a and b, are on the adiabatic curve, $a\,b$, we have the following relation between the co-ordinates of these points—

$$p_1 v_1^n = p_2 v_2^n, \quad . \quad . \quad . \quad (1).$$

n being a positive quantity, greater than unity (see Lecture XV.) Similarly, for the adiabatic curve, $c\,d$, we have

$$p_3 v_3^n = p_4 v_4^n, \quad . \quad . \quad . \quad (2).$$

Also, for the isothermal curves, $b\,c$ and $a\,d$, we have the two equations—

$$p_2 v_2 = p_3 v_3, \quad . \quad . \quad . \quad . \quad (3).$$

$$p_1 v_1 = p_4 v_4. \quad . \quad . \quad . \quad . \quad (4).$$

We have thus four equations, from which, if we eliminate the four pressures, we obtain the relation

$$\frac{v_3}{v_2} = \frac{v_4}{v_1}.$$

This enables us to simplify the expression for the work done, which now becomes

$$W = J\,H\left\{1 - \frac{\tau_1}{\tau_2}\right\}$$
$$= J\,H\left\{\frac{\tau_2 - \tau_1}{\tau_2}\right\}.$$

And the efficiency of the engine

$$= \frac{W}{J\,H} = \frac{\tau_2 - \tau_1}{\tau_2}.$$

For example, suppose that the air in our ideal engine is raised to a temperature of 400°, and, after doing work, its temperature is 32° Fah.;

$$\text{Then,} \quad \tau_2 = 400 + 460 = 860$$
$$\text{,,} \quad \tau_1 = 32 + 460 = 492$$

$$\text{Work done} = J\,H\left(\frac{\tau_2 - \tau_1}{\tau_2}\right) = J\,H\left(\frac{860 - 492}{860}\right) = \cdot 43\,J\,H\text{, nearly.}$$

Again, returning to the equation,

$$\frac{v_3}{v_2} = \frac{v_4}{v_1},$$

which may be written in this form—

$$\frac{Og}{Oe} = \frac{Oh}{Of},$$

or, $$Og : Oe :: Oh : Of,$$

we have the relation which must hold between the volumes of the substances at the various stages of the cyclical process just described. This relation enables us to determine the value of Og, so that the adiabatic expansion from position 3 to 4 will cause the temperature to fall to τ_1°.

Oe, Of, and Oh are supposed to be known, hence a simple calculation will give us Og, or the abscissa of the point 3, because

$$Og = \frac{Oe\,.\,Oh}{Of}.$$

Carnot's Principle.—The following important principle was laid down by Sadi Carnot, the founder of the theory just given, in 1824:—

The amount of work done by a reversible heat engine depends *only* on the constant temperature at which heat is received, and at which it is rejected, and is independent of the nature of the intermediary agent (such as steam, air, &c.) Its efficiency is consequently a maximum.

We see, therefore, that the amount of work got out of a heat engine, depends entirely on the absolute temperatures between which it is worked. In order to obtain the whole of the work from a mass of heated air, it would be necessary to cool it down to the absolute zero—a process beyond the reach of practice; and, hence, our imaginary engine, which is absolutely perfect in its action, is only able to yield a *portion* of the energy stored in the gas in the form of heat. The example worked out shows this clearly, for our perfect engine, in working between the temperatures 400° and 32°, can only convert ·43 of the heat energy into actual mechanical work. The efficiency of any heat engine used in actual practice, such as a steam or an air or a gas engine, is considerably less than this.

Entropy and Thermodynamics from an Engineer's Point of View.—Although a few lectures have been written by the author for this book upon this subject, in order to meet the modern requirements of engineering examinations, as well as to place before the independent student the latest views on the same, we must confess, that the suddenness with which the present edition was called for has prevented these lectures being got ready in time for this issue. Our best plan, under these circumstances, is to herewith recommend a few of the latest books and articles on this important subject, which naturally follows the consideration of "Carnot's Principle":—1896, "The Thermal Efficiency of Steam Engines," by H. R. Sankey, Captain R.E., M.Inst.C.E. (of Messrs. Willans and Robinson, Rugby); see *Trans. Inst. C.E.*, vol. xxv., p. 182, &c.; 1898, *The Theta-Phi Diagram applied to Steam, Gas, Oil, and Air Engines*, by H. A. Golding, A.M.I.M.E., published by The Technical Publishing Coy., Manchester; 1900, "Work and Heat, Entropy, Water Steam, $\theta\phi$ Diagrams," see Chapters XXI. to XXIII. in Prof. Perry's *Text-Book on the Steam Engine*; 1903, *Thermodynamics of Heat Engines*, by Prof. S. A. Reeve, Worcester Polytechnic, U.S.A., published by Macmillan & Co., London and New York; 1903, *Treatise on Thermodynamics*, by Dr. Max Planck, Translation, published by Longmans, Green & Co., London; 1903-04, Four articles in *Engineering* from August 28 to September 18, 1903, on "Entropy or Thermodynamics from an Engineer's Standpoint," by J. Swinburne, ex-President of the Institution of Electrical Engineers; also, Book on same, published by Archibald Constable & Co., 1904.

Lecture XIII.—Questions.

1. Having regard to the theory of heat, will you state some reasons for concluding that when steam expands in a cylinder behind a working piston, the law of expansion differs from that of Boyle?

*2. Define a heat engine. State the conditions under which such an engine will give out the greatest quantity of work, and establish your statement by reasoning. A perfect heat engine receives heat at 350° F., and rejects heat at a temperature of 90° F. : Find its efficiency. *Ans.* = ·32.

3. An engine uses 10 lbs. of steam per minute, the feed temperature is 60° F., the boiler temperature 300° F., and that of the condenser 104° F., what is the theoretical maximum efficiency of the engine? State Regnault's formula for the total heat of steam at a given temperature, and deduce the amount of heat which each pound of steam has received in the boiler. What horse-power would be developed if the engine worked as a perfect engine? *Ans.* ·258 ; 1144·4 ; 69 H.P.

4. Find an expression for the efficiency of an elementary heat engine.

5. Investigate a method of ascertaining the absolute temperature which corresponds to 100° F.

6. What is meant by the adiabatic expansion of a gas? If you were required to set out approximately the curve of adiabatic expansion of steam, how would you proceed?

7. A steam engine at the mouth of a coal pit is employed to compress air to a pressure of 3 atmospheres. The air becomes heated, but is cooled down by water to a temperature of 100° F. It is then conveyed in a pipe to the bottom of the pit and to some distance along its workings, being finally caused to drive the working piston of an ordinary high-pressure engine. The air when liberated produces a freezing temperature in its neighbourhood. Apply your knowledge to explain this fact.

8. What do you understand by the term "heat engine"? Define the efficiency of a heat engine and show why it is that only a small proportion of the heat absorbed by a heat engine reappears in the form of useful work, and show also which of the sources of loss must occur even in a perfect heat engine. If an engine and boiler consume 3 lbs. of coal per hour per horse-power and the heat developed during the combustion of each lb. of coal is sufficient to convert $12\frac{1}{2}$ lbs. of water at 62° F. into steam at an absolute pressure of 100 lbs. per square inch (temperature, 327·9° F.), what, under these circumstances, is the efficiency of the engine, the boiler efficiency being taken as 72 per cent.? (S. & A., 1897, Adv.)

9. Write out the formula enabling us to calculate the heat which must be given to stuff when (1) at constant volume, v_1, the pressure changes from p_1 to p_2 ; (2) at constant pressure, p_1, the volume changes from v_1 to v_2 ; (3) according to any way of altering in pressure and volume. Prove these rules from first principles, and from the two laws of thermodynamics. (S. & A., 1897, Hons.)

10. Calculate the work done per lb. of steam by a *perfect steam* engine working between the absolute temperatures, t_1 and t_2. That is—1 lb. of water is heated as water from t_1 to t_2, and is then converted into steam, expanded adiabatically till it reaches t_1, and the operation completed along an isothermal to the starting point. Show why Mr. Willans found this a good enough standard for non-condensing, but unsatisfactory for condensing engines. (S. & A., 1897, Hons.)

11. What is the law connecting the pressure, volume, and absolute temperature of 1 lb. of air? Consult the printed table furnished you at the

end of Text-book for the density of air. Why is the specific heat greater at constant pressure than at constant volume? (B. of E., 1900, Adv.)

12. Steam enters a cylinder at 150 lbs. (absolute) per square inch. It is cut off at one-fourth of the stroke, and expands according to the law "pv constant." Find the average pressure (absolute) in the forward stroke. If the back pressure is 17 lbs (absolute) per square inch, what is the average effective pressure? If the area of the cross section of the cylinder is 126 square inches, and the crank is 11 inches long, what work is done in one stroke? Neglecting clearance and condensation, what volume of steam enters the cylinder per stroke? If the admitted steam has a volume of 3 cubic feet to the lb., what is the weight of steam admitted per stroke? What work is done per lb. of steam? (B. of E., 1901, Adv.)

13. Steam enters a cylinder at the absolute pressure, 120 lbs. per square inch, and expands according to the law "pv constant" Neglect clearance and cushioning, and use the ordinary hypothetical diagram. Constant back pressure, 27 lbs per square inch. Take the following values of the cut-off:—Half-stroke, quarter-stroke, eighth of stroke. Find in each case the effective pressure The area of the piston is 1 square foot: stroke, 2 feet; what is the work done per stroke? What is the work done per cubic foot of steam entering the cylinder? Tabulate your answers. (B. of E., 1903, Adv.)

14. What is the law connecting pressure, volume, and temperature of 1 lb. of air, if at 1 atmosphere and 0° C the volume is 12·39 cubic feet? At 2½ atmospheres and 130° C., what is its volume? It receives heat energy equivalent to 300,000 foot-pounds at constant *volume*. What are its new pressure and temperature? The specific heat of air at constant *pressure* is 0·238. (B. of E., 1903, Adv. and H., Part I.)

LECTURE XIV.

CONTENTS.—Distribution of Steam in a Cylinder—Lap and Lead of a Valve, &c., Angle of advance of an Eccentric, Points of admission, Cut-off, Release, and Compression—Diagram of the relative positions of Crank and Piston—Zeuner's Valve Diagrams.—Questions.

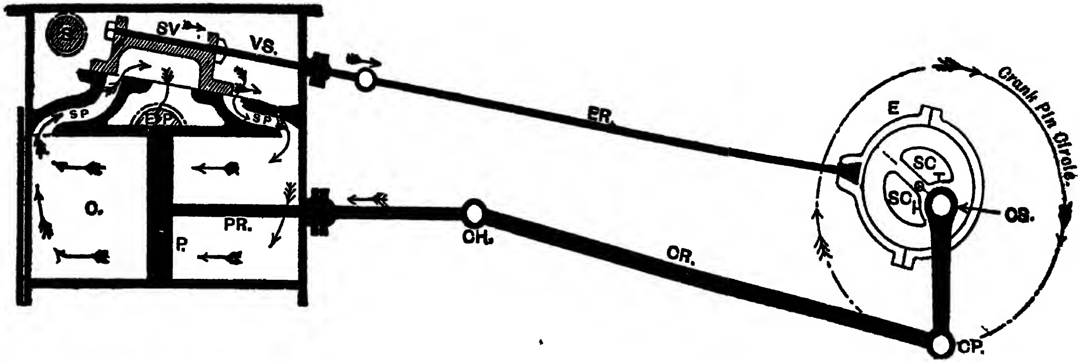
Distribution of Steam in a Cylinder.—Before explaining the Indicator, and the results obtained by it in the form of Indicator Diagrams from various types of engines, it will be necessary to describe generally the action of the eccentric, slide valve, crank, connecting rod, and piston, with their relative positions, so as to understand the distribution of steam in the cylinder of an ordinary engine.

This is most graphically and easily done by aid of a large skeleton working model, in which the slide valve, piston, &c., are all shown in section in one plane.

By aid of this model (fitted with a set of slide valves having different dimensions, the means of fixing the eccentric at different angles to the crank, and of altering the link motion or the travel of the slide valve), the distribution of steam in a cylinder may be studied simultaneously by a large class.

The engine, as seen by the arrow on the crank pin circle, is going ahead or turning with the hands of a watch. The valve is moving forward and on the point of cutting off steam from the cylinder at about half-stroke, while the piston is moving back towards the after end of the cylinder, and will therefore complete the rest of the stroke under the expansion of the steam.

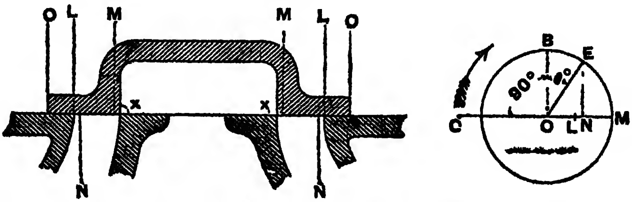
It will be seen from the figure that the eccentric, E, consists of a simple pulley placed eccentrically $2\frac{1}{4}$ inches to the crank shaft centre, *i.e.*, with a throw of $2\frac{1}{4}$ in. The to and fro movement of the slide valve is obtained from this eccentric through the intermediate mechanism of the eccentric strap, eccentric rod, E R, and the valve spindle, V S, while the to and fro movement of the piston, P, is obtained from the crank keyed to the crank shaft, through the intermediate mechanism of the connecting rod, C R, and the piston rod, P R. It is, therefore, clear that, in order to ascertain the effect of the slide valve in distributing steam to the cylinder, its position at any point of the piston's stroke must be studied by observing the relative positions of the eccentric and the crank.



C for Cylinder.	C P for Crank pin.	S V for Slide valve.	S C for Set screw for adjusting eccentric to any angle with respect to the crank.
P ,, Piston.	C S ,, Crank shaft.	V S ,, Valve spindle.	
P R ,, Piston rod.	S ,, Steam pipe.	E R ,, Eccentric rod.	
C H ,, Cross head.	S P ,, Steam ports.	E ,, Eccentric pulley.	
C R ,, Connecting rod.	E P ,, Exhaust ,,		

The above diagram represents a working model belonging to the College of Science and Arts, Glasgow, about 10 feet long, with a cylinder 18 inches diameter and 24 inches stroke, presented by the President, David Rowan, Esq., M. Inst. C.E., and fitted in the College workshop by one of the students, Mr. James Welsh, with a separate set of link motion, double eccentrics, &c., which can be put on in a few minutes.

Lap and Lead of a Valve, &c.—The slide valve shown in the following figure is purposely placed at the centre of its stroke, in order to facilitate an explanation of what is meant by *lap*. The valve consists of a hollow box with projecting ends, the lower face being accurately planed and fitted, so as to be steam tight on the valve port face. The hollow arch of the valve just covers the distance between the inner edges of the steam ports, so that the moment the valve cuts off the exhaust from one end of the cylinder it opens the other end of the cylinder to exhaust. Sometimes, however, slide valves have what is termed *inside lap*, that is, an inner projection at each end of the arch of the valve, marked in dotted lines by, *x*, in the figure. This causes the exhaust to take place later on the one side and to be cut off sooner on the other side of the piston. The effect of this is twofold—(1) a later release causing a higher back pressure, (2) compression before the end of the stroke. The latter result is useful, as we shall see in the next lecture, owing to arresting the momentum of the moving piston, piston-rod, crosshead and connecting-rod, and thus lessening what would otherwise be a sudden stress or jerk on the cross-head and crank-pin bearings, causing undue wear and tear. This is termed compression or cushioning; frequently however, part of the cushioning is effected by giving "*lead*" to the slide valve, that is, allowing it to open the steam port before the piston has come to the end of its stroke.



Now, looking at the left-hand figure we see the three dotted vertical lines drawn above the valve face at each end of the valve. The distance, O to L, is the amount by which the valve overlaps the steam port at each end. This is termed the *outside lap* of the valve, while the distance between L and M is the amount the valve (when at the end of its stroke) opens the steam port for admission of steam into the cylinder. This distance, L M, is frequently less than the breadth of the steam port, because the same passage serves both for inlet of the steam to, and its exit from, the cylinder; and, seeing that the steam has expanded in bulk while doing work in the cylinder, the freer and the quicker the exhaust, the less will be the back or obstructive pressure.

The vertical dotted lines drawn from, N, below the valve face near the outside edge of each steam port, indicate the *lead*.

The circle in the right-hand figure is taken to represent the path of the centre of the eccentric pulley, which works the slide valve. The radius, O M, is, therefore, equal to the throw of the eccentric, or half the travel of the valve. Now, supposing the crank to be in the position, O C, or level at the inner dead centre in a horizontal engine (*i.e.*, the piston is just at the commencement of the outgoing stroke), mark off the distance, O N, equal to the outside lap, O L, plus the lead, L N, draw N E perpendicular to O M, and join O E; then (neglecting the obliquity of eccentric rod) we have—

O C for the Centre line of the crank.	L M for the Maximum opening to steam
O E ,, ,, ,, eccentric.	$\theta°$ for the Angle, B O E, or the angle of advance.
O L ,, Lap.	
O N ,, Lap + lead.	

We thus see that the centre line of the eccentric must be in advance of the centre line of the crank, by $(90° + \theta°)$, where $\theta°$ is called the *angle of advance*.

If there was neither lap nor lead, then the centre line of the eccentric would be at right angles to the centre line of the crank, or the eccentric only 90° ahead of the crank.

In order to impress these various parts and positions of the slide valve, we again enumerate them as definitions.

Lap or cover of a slide valve is the amount by which the edge of the valve overlaps the adjoining edge of the steam port, when the valve is in the middle of its stroke, and is termed *outside* or *inside* lap, according as we refer to the outside or inside of valve.

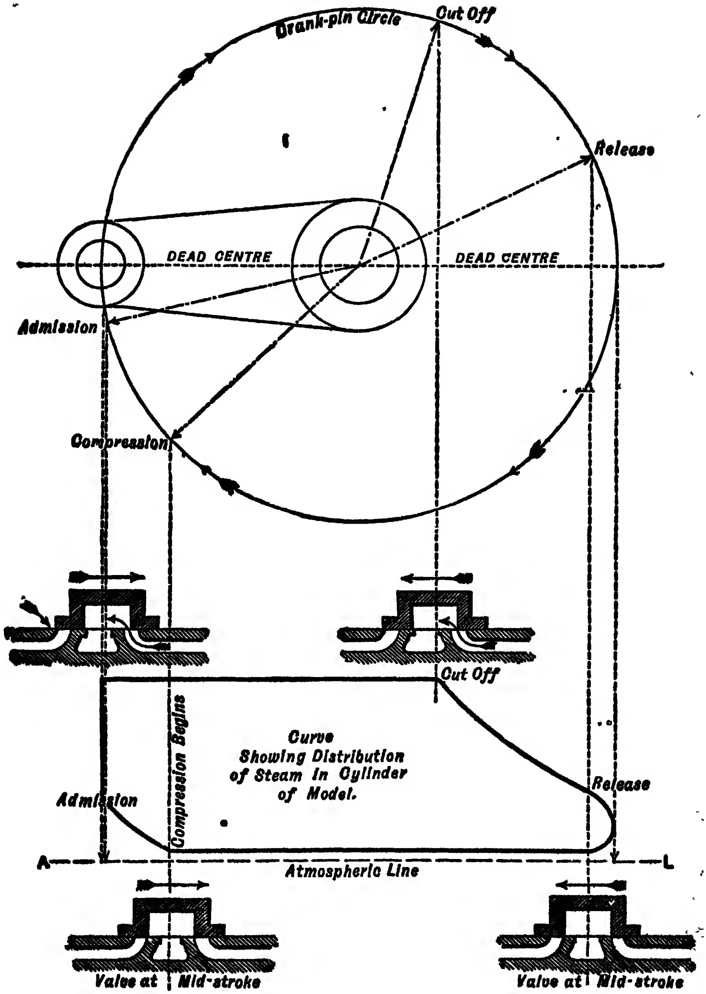
Lead is the amount of the opening of the steam port at the beginning of the piston's stroke.

Angle of advance of eccentric is the angle by which the centre line of the eccentric stands in advance of that position, which would bring the valve to its mid-stroke when the crank is on the dead point; or, in other words, the angle between the crank and the centre line of the eccentric *minus* 90°.

* *The throw of an eccentric* is the distance between the centre of crank shaft and the centre of eccentric pulley.

The travel of a slide valve is equal to the distance the valve moves to and fro in one stroke of the piston, or twice (the lap + opening to steam). It is equal to *twice the throw* of eccentric.

* Several authors—*e.g.*, Prof. Goodeve—state that *the throw of an eccentric* is equal to the diameter of the circle described by the centre of the eccentric pulley. The word throw is ambiguous, and might be discarded, for it is liable to lead to confusion. See *The Practical Engineer*, Nov. 18, 1887, p. 521.

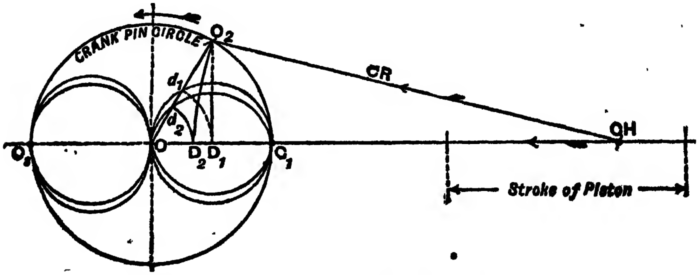


RELATIVE POSITIONS OF CRANK AND SLIDE-VALVE, WITH CURVE SHOWING THE DISTRIBUTION OF STEAM IN THE CYLINDER.

The foregoing diagram illustrates the four principal points in the motion of the simple D slide valve of the working model explained at the beginning of this lecture, p. 170, as well as the corresponding positions of the crank, and also the probable distribution of steam in the cylinder or "diagram of work."

1. The point of *admission* of steam to the cylinder.
2. The point of *cut-off* of steam from the cylinder.
3. The point of *release*, or when exhaust begins.
4. The point of *compression*, or when exhaust stops.

The diagram is self-explanatory, in as far as it shows how each of these points marked on the crank pin circle are projected on to the "diagram of work" (or piston's stroke) below, with the corresponding positions of the slide valve sketched on the lines of projections. The direction of motion of the crank and of the slide valve at each point is also made clear by arrows. It will be observed that the slide valve is at the same position with respect to the steam ports when beginning to admit steam to the cylinder and to cut off the supply of steam from the same, and that its direction of motion is in each case opposite to the direction of the piston's motion. It is also evident that the slide valve is at the middle of its stroke when release and when compression begins, and that its motion is opposite in each ease to that of the piston's motion, as indicated by the straight arrows placed directly above the slide valve.



Relative Positions of the Crank and the Piston.—The following method of determining relative positions of the crank and the piston is of great importance, because it is the method used in determining the relative positions of the slide valve and its eccentric.

In the fig., let O O_2 represent the crank, then with centre, O_2, and radius (C R) = the connecting-rod, describe an arc cutting the centre line of the engine's stroke in (C H), which gives the position of the crosshead. With this point (C H) as a centre, and the

length of (O R) as radius, describe the arc, $C_2 D_2$. The length $O D_2$ will be equal to the distance of the piston from the *middle* of its stroke when the crank is in the position $O C_2$. If this distance, $O D_2$, be set off along the crank, $O C_2$, by drawing with centre, O, and radius, $O D_2$, the arc, $D_2 d_2$, and the same operation be repeated for a series of different positions of the crank, all these points will be found to lie on the polar curve, $O d_2 C_1$. Any chord of this curve drawn from the point O will be equal to the distance of the piston from the middle of its stroke when the crank lies along that chord.

The double looped curves in full lines are the curves obtained by this method.

If the connecting-rod be infinitely long it is evident that instead of the arc, $C_2 D_2$, we get the straight line, $C_2 D_1$, at right angles to the line of stroke, and that $O D_1$ is, in this case, the distance of the piston from the middle of its stroke. If this distance be set off along the crank by drawing the arc, $D_1 d_1$, and the same operation be repeated for a series of different positions of the crank, it will be found that all these points lie on a pair of circles drawn with $O C_1$ and $O C_3$ as diameters. These are shown by the dotted circles in the figure.

The effect of the obliquity of the connecting-rod is well seen by comparing the curves in full lines with the circles in dotted lines.

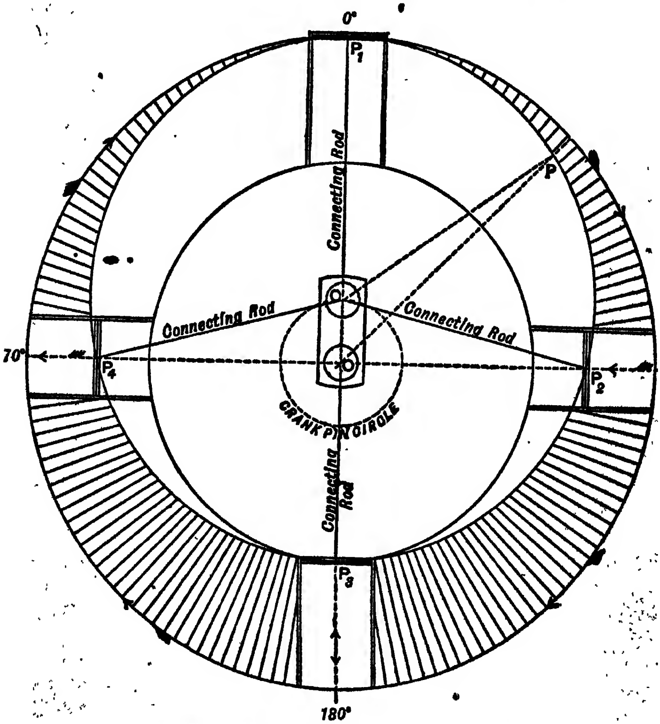
In valve diagrams it is usual to neglect the effect of the obliquity of the eccentric-rod, because the ratio of its length to the throw of the eccentric is generally great, and its effect is therefore generally not worth taking into account.

Another method of finding the relative position of the crank and the piston is as follows:—(see next fig.)

Draw two lines, $P_1 P_3$, $P_2 P_4$, at right angles to each other. From their intersection, O, with radius O C, equal to the length of the crank, describe the inner or crank pin circle. With O as a centre and $O P_1$ as radius equal to the length of the connecting rod, describe the circle $P_1 P_2 P_3 P_4$. With O as a centre and $C P_3$ as a radius, describe a circle; again, with the centre, O, and radius, $O P_1$, describe the large outer circle.

Now, suppose the crank to be fixed in the position, O C, and the cylinder to revolve round the centre, O, in the direction of the hands of a watch, as indicated by the arrows, then the connecting-rod (which is supposed for the purposes of explanation to be connected directly to the piston) will cause the piston to move inwards during the first half, and outwards during the second half of the revolution. The positions P_1, P_2, P_3, and P_4 represent the piston at 0°, 90°, 180°, and 270° of the revolution. For any

required angle, O C P, between the crank, O C, and the connecting-rod, C P, the position of the piston is indicated by the position, P. The radial lines between the large outer circle and the inner circle of radius, O P_1, indicate the distance of the piston from the outer end of the cylinder; for the path of the piston lies in the line of the circle P_1, P_2, P_3, P_4. Precisely the same



RELATIVE POSITIONS OF CRANK AND PISTON.

reasoning will hold good if we consider the crank to revolve and the cylinder to be fixed (as is usually the case) say in the position at P_1. Then the radial lengths between the outer circle and the circle P_1, P_2, P_3, P_4 respectively represent the distance of the piston from outer end of the stroke for each position of the crank during a revolution.

Cause of the Unequal Distribution of Steam during the Forward and the Back Stroke of the Piston.—If the connecting-rod of an engine was infinitely long (and therefore remained always parallel to the centre line of the piston's motion), the point of "cut off," and consequently the distribution of steam, would be equal at both ends of the cylinder; but when the length of the connecting-rod (as usually adopted in practice) is only from 2 to 4 times the length of the crank, the distance to the point of "cut off" is considerably later on the forward stroke than on the return or back stroke.

An explanation of the following two diagrams will render this quite evident.

Firstly. Consider a case when the slide valve has "outside lap" only, and no "lead."

1. Draw a centre line of the piston's motion, C_3, to 10.

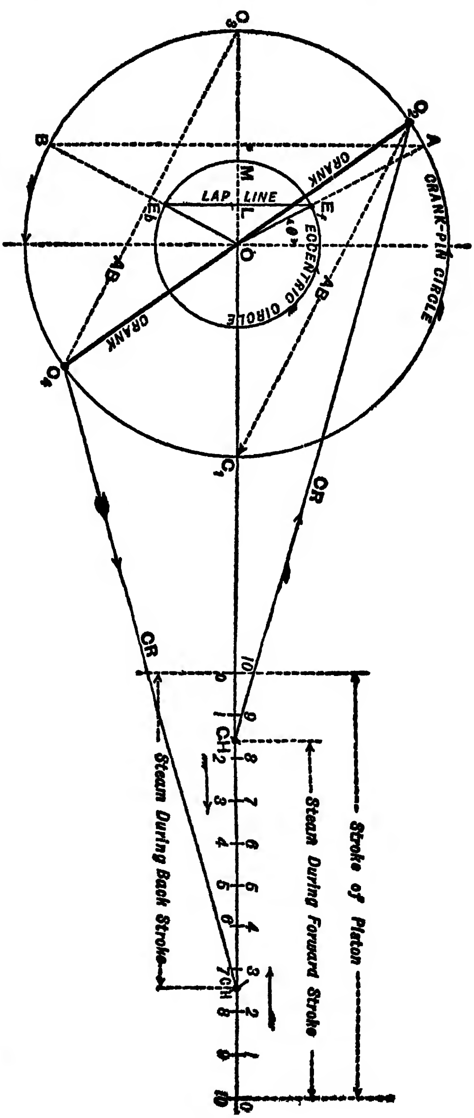
2. With any convenient position, O, as a centre and radius, $O\,C_1$, equal to the length of the crank, describe a circle, $C_1C_2C_3C_4$, and let the crank revolve in the direction shown by the arrows on the crank pin circle.

3. With centre C_1, and radius equal to the length of connecting-rod (in this case = 3 cranks), describe an arc, cutting the centre line of engine in the position 10 furthest from C_1. With centre, C_3, and the same radius, describe another arc, cutting the same centre line in position 10 nearest to C_1. Then the distance, 0 to 10, is equal to the piston's stroke. This distance may be conveniently divided into ten equal parts both above and below the centre line, so as to indicate percentages of the stroke during the *forward* and *back* strokes of the piston.*

4. With centre, O, and radius equal to the throw (or eccentricity of the eccentric), describe the inner small circle $M\,E_f\,E_b$.

5. From O plot off a distance, O L, equal to the outside lap of the slide valve, and draw through, L, the line, $E_f\,L\,E_b$, at right angles to the centre line of the engine. From O, draw radial lines, $O\,E_f\,A$ and $O\,E_b\,B$, cutting the crank pin circle in A and B, and join A B. Then, since the slide valve has no "lead," $O\,E_f$ is the centre line of the eccentric when the crank is in the position $O\,C_1$, and the eccentric turns round from the position $O\,E_f$ to the position $O\,E_b$, in the operation of moving the slide valve during opening and closing the back steam port; consequently, the crank must turn through an equal angle during

* Of course the positions 10 and 10 are in reality the centre of the cross-head at each end of the stroke in ordinary engines having a crank and connecting-rod. To include the length of the piston-rod would extend the figure beyond the limits of the page.



UNEQUAL DISTRIBUTION OF STEAM DURING THE FORWARD AND BACK STROKES OF THE PISTON, DUE TO THE LENGTH OF THE CONNECTING-ROD, WHEN THE SLIDE-VALVE HAS ONLY OUTSIDE LAP.

this operation, *i.e.*, an angle equal to A O B. The "angle of advance" of the eccentric is indicated by $< \theta >$.

6. With centre, C_1, and radius, A B, describe an arc, cutting the crank-pin circle in C_2, and join O and C_2 by a thick line. Then, O C_2 is the position of the crank when steam is cut off from the cylinder, *i.e.*, when the centre of the eccentric pulley is in the position E_2.

7. With C_2 as a centre and radius equal to the length of the connecting-rod, describe an arc cutting the centre line of the engine in C H (cross-head), nearly midway between positions 8 and 9, above the line, thus showing that steam is cut off at about 85 per cent. of the stroke during the forward movement of the piston and crank.

8. With C_3 as a centre and radius, A B, describe an arc, cutting the crank-pin circle in C_4, and join O and C_4 by a thick line. Then O C_4 is the position of the crank when steam is cut off from the cylinder during the back stroke of the piston.

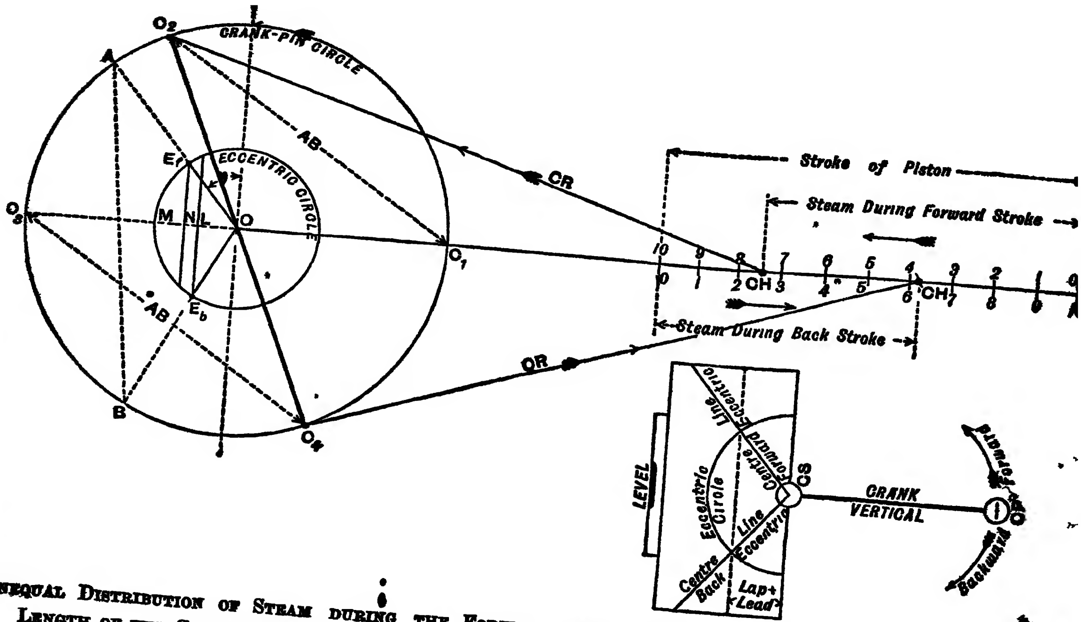
9. With C_4 as a centre and radius equal to the length of the connecting-rod describe an arc cutting the centre line of the engine in C H, nearly midway between positions 7 and 8, below the line, thus showing that steam is cut off at about 75 per cent. of the stroke during the backward or return movement of the piston and crank.

Secondly. Consider a similar case in every respect to the last, except with "lead" as well as "outside lap" given to the slide-valve.

Perform precisely the same construction as before, with this addition, viz., that O N is equal to the lap O L, plus the lead L N; consequently, O E_1 is the centre line of the eccentric when the crank is in the position O C_1, and O E_5, the centre line of the eccentric when steam is cut off during the forward stroke. The eccentric therefore turns round through the angle A O B while the crank turns from O C_1 to O C_2, and the piston moves from position O to nearly midway between 7 and 8 above the centre line of engine, as indicated by the letter C H.

On the return or back stroke the crank turns from O C_3 to O C_4, while the piston moves from O to a little beyond figure 6, below the centre line. Steam is therefore cut off at about 75 per cent. during the forward stroke, and 62 per cent. during the back stroke, as against 85 and 75 per cent. when no lead was given to the slide-valve.

Fixing Forward and Backward Eccentrics.—In large marine, locomotive, and other engines, the "forward" and back driving eccentrics are often fixed in their permanent positions on the crank shaft before the crank is fitted into its bearings. The method by which their proper position is ascertained relatively



UNEQUAL DISTRIBUTION OF STEAM DURING THE FORWARD AND BACK STROKES OF THE PISTON, DUE TO THE LENGTH OF THE CONNECTING-ROD WHEN THE SLIDE-VALVE HAS LAP AND LEAD. ALSO A METHOD OF FIXING THE POSITION OF THE ECCENTRICS BEFORE FITTING AN ENGINE TOGETHER.

to their own crank will be readily understood from the foregoing diagrams and explanation, and by also considering the small figure to the left hand side of the last diagram.

Fix the crank in a vertical position, and place on the crank-shaft a wooden or sheet-iron template, with the upper edge level, having previously drawn upon it the centre lines of the "forward" and "backing" eccentrics, by the rule just described for lap and lead. Now mark on the crank-shaft with a Λ box-square the centre lines of the feathers for fixing each of the eccentrics, and line off the outline of these feathers where they are to be sunk (parallel to the crank-shaft). Previous to placing the template on the crank-shaft, a semicircular hole has of course been cut from it to fit the crank-shaft. This handy method saves all the time, trouble, and expense of temporarily putting the crank-shaft in its bearings and fitting together the connecting-rod, piston-rod, piston, eccentric straps, eccentric rods, and slide-valve, then turning the whole engine and ascertaining by trial the best position of the eccentrics (which was quite commonly done until a few years ago), and then disconnecting the whole in order to get the feathers sunk in the crank-shaft for keying on the eccentrics in the positions that had been ascertained by trial and observation.

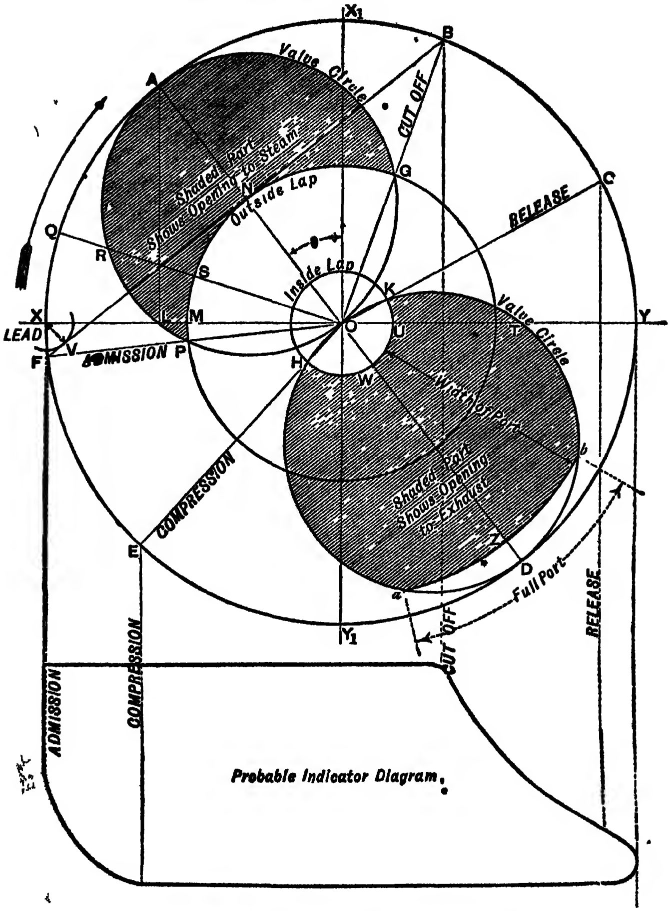
Zeuner's Diagram of Simple Valve Motion.—Of the problems relating to the motion of a slide-valve, the case most commonly recurring in practice, is that in which we have given the position of the crank at the point of cut-off, the travel of the valve, and the amount of lead, and have to determine the angular advance of the eccentric and the amount of outside lap required.

With centre O (see next figure) and radius equal to the throw of eccentric or half-travel of the valve, describe the circle A B C D E, and through O, draw X Y and X_1 Y_1 at right angles to each other. From O, draw O B, to represent the position of the crank when the steam is to be cut off; this is determined by means of the previous diagrams. With X as centre, and radius X V, equal to the lead of the valve, describe part of a circle, and from B, draw B F, touching this circle. Through O, draw A D at right angles to B F.

On O A* and O D, describe the valve circles A L G and H K D. These are sometimes termed the "primary" and "secondary" valve circles respectively. With O as centre, describe the circle P M N G, touching the line B F at the point N. This circle is

* The line O A would be the centre line of an eccentric for the crank in the position O Y, *and going backwards* or opposite to the direction of the arrow in the figure; while the angle A O X_1, or θ, would be the *angle of advance* for that eccentric.

known as the *outside lap circle*, O M being equal to the outside lap, and M L equal to the lead.



ZEUNER'S DIAGRAM OF VALVE MOTION.

If the position of the crank at the point of compression or the point of release is given, draw O E or O C to represent one of these, and where this line cuts the valve circle, H K D, describe

a circle with centre O to pass through that point. The radius of this circle gives the amount of the *inside lap.*

Having now completed our diagram, we can see the distance the valve has moved from its central position for any position of the crank, and also the opening of the port to steam at that point. Suppose the crank to be in the position O Q, and moving in the direction of the arrow, then the distance which the valve has moved from its central position is given by O R (or that portion of the line which is included within the valve circle A L G); and since the outside lap of the valve is equal to O S, therefore the opening of the port to steam is equal to R S. When the crank reaches the position O A, the port has its maximum opening equal to A N. As it passes position O A, the valve begins to close the steam port; and when it arrives at O B, the steam port is closed altogether and the steam cut off. Therefore we see that, when the crank is in the position O F, the valve is just beginning to open the steam port; and when it reaches the dead point O X, the steam port is open an amount = L M = X V = the lead of the valve, which equality is easily proved by geometry.

When the crank reaches the position O C, the valve has passed its middle position, and is distant from it on the other side an amount equal to O K; and as this is equal to the inside lap of the valve, therefore the exhaust port opens, and release takes place at this point. As the crank passes the position O C, the valve continues to open the port to exhaust. Thus, when the crank arrives at O Y, the valve has moved from its central position a distance equal to O T; and, since O U is the inside lap, therefore the port is open to exhaust an amount equal to O T − O U = U T. If W Z represents the width of the port, it is evident that, when the crank reaches the position O D, the valve has travelled beyond the port a distance equal to Z D, and, therefore, if the arc *ab* be drawn through Z, it is apparent that, during the motion of the crank from *b* to *a*, the port remains full open to exhaust. When the crank comes to the position O E the port is completely closed to exhaust; and, since the piston is not yet at the end of its stroke, compression takes place in the cylinder.

The shaded part of the upper or primary valve circle, represents the opening of the port to steam for different positions of the crank; and when the line representing the position of the crank cuts the shaded part, it indicates that the port is open to steam, by an amount equal to that portion of the line included between the two bounding curves of the shaded part. Similarly, the shaded part of the lower or secondary valve circle represents the opening of the port to exhaust, and, as we have seen, it is full open during the passage of the crank from *b* to *a*, it is not

usual to open the ports fully *to steam*, as explained at the beginning of this Lecture, hence no line corresponding to $a\,b$ appears in the upper valve circle.

The student should work out a few examples, in order to impress the construction on his memory; for, if once the principle of the diagram is fully grasped, no difficulty will be found with any of the various problems relating to the motion of the slide valve.

For example, given *the travel, the lap, and the angle of advance*, to find the point of cut-off, the lead, etc., draw the circle, A B C D E, to represent the travel, as before; also the outside lap circle, P M N G, and making the angle θ equal to the given angle of advance, draw A D to represent the centre line of the valve circles. Describe the valve circles as before, then we see that when the crank is in the position O X, the valve is open an amount equal to L M, therefore L M is the lead of the valve. Through the point G, where the valve circle cuts the lap circle, draw O B, then O B is the position of the crank at the point of cut-off.

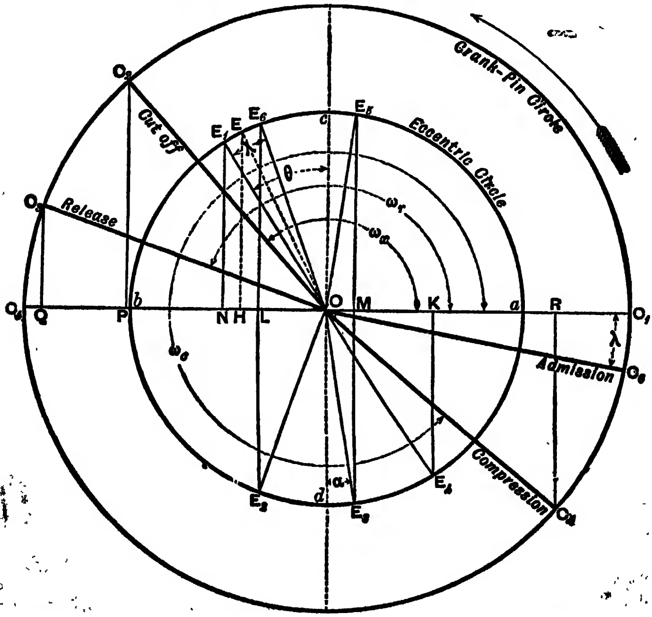
Or, suppose we are given *the travel, the lap, and the lead*, and are required to construct the diagram and to find the angle of advance and the point of cut-off. Having drawn the circle, A B C D E, representing the travel, lay off O M equal to the lap, and M L equal to the lead of the valve. From L, draw L A perpendicular to X Y; then the angle, A O X_1, is the angle of advance, viz., θ, required. With centre, O, and radius, O M, describe the outside lap circle, and on A O describe the valve circle as before. These circles intersect in G Through G draw O B, then O B is the position of the crank at the point of cut-off.

Under the diagram of the valve motion we have given the probable indicator diagram, showing the admission, cut-off, release, and compression, taking place at the proper points of the stroke.

Formulæ for Ordinary Slide Valves.—In the remaining portion of this Lecture we purpose finding algebraical expressions for the more important relations connecting the various quantities in the ordinary slide valve. We have already shown how, on obtaining sufficient data, these quantities can be readily found by means of a Zeuner's valve diagram. This latter method, although very instructive and accurate, requires the use of a drawing board and drawing instruments. These are not always at hand; hence, for the benefit of those engineers and students who can manipulate simple algebraical and trigonometrical formulæ, we shall deduce by mathematics the more important relations connecting together the "lap," "lead," "travel of valve," "cut off," &c.

In what follows :—

Let T represent the travel of valve in inches.
„ L_o „ "outside lap" „
„ L_i „ "inside lap" „
„ l_o „ "outside lead" „
„ l_i „ "inside lead" „
„ S „ stroke of piston in feet.
„ x „ distance from beginning of stroke to point of cut off in feet.
„ θ = angle of advance of eccentric.
„ λ = angle of lead—*i.e.*, angle which centre line of crank makes with centre line of engine when steam is just being admitted to cylinder.
„ ω_s = angle that crank moves through from beginning of stroke to point of cut off.
„ ω_r = angle that crank moves through from beginning of stroke to point of release of steam.
„ ω_c = angle that crank moves through from beginning of stroke to point where compression begins.



Referring to the figure where the larger circle, $C_1 C_2 C_5$, represents the crank pin circle, and the smaller circle, $a\, c\, b\, d$, the eccentric circle exaggerated, $a\,b = T$, $O\,L = L_o$, $L\,N = l_o$.

$$\angle c\,O\,E_1 = \theta, \therefore \angle E_1\,O\,E_6 = \angle C_1\,O\,C_6$$

Now, we have already seen, in this lecture, that E_1 is the position of the centre of the eccentric when the piston is at the beginning of its stroke, and E_2 is the position of the centre of the eccentric when the steam is cut off. Let C_2 be the position of crank pin when cut off takes place, then—

$$\angle E_1 O E_2 = \angle C_1 O C_2 = \omega_x.$$

$$\text{Also } \angle c O E_1 + \angle E_1 O E_2 + \angle E_2 O d = 180^\circ;$$

$$\text{But } \angle E_2 O d = \angle c O E_6 = \theta - \lambda,$$

$$\therefore \theta + \omega_x + \theta - \lambda = 180^\circ,$$

$$\therefore 2\theta + \omega_x - \lambda = 180^\circ \quad . \quad . \quad . \quad . \quad (1).$$

$$\text{Now } \frac{O N}{O E_1} = \cos \angle E_1 O N = \sin \angle c O E_1 = \sin \theta,$$

$$\therefore \frac{L_o + l_o}{\frac{1}{2} T} = \sin \theta, \; i.e., \; \frac{2(L_o + l_o)}{T} = \sin \theta, \quad . \quad (2).$$

$$\text{Similarly } \frac{O L}{O E_6} = \sin(\theta - \lambda), \; \therefore \frac{2 L_o}{T} = \sin(\theta - \lambda), \quad . \quad (3).$$

But, from equation (1) we get—

$$\theta - \lambda = 180^\circ - (\theta + \omega_x), \; \therefore \sin(\theta - \lambda) = \sin(\theta + \omega_x).$$

Substituting this in equation (3) we have—

$$\frac{2 L_o}{T} = \sin(\theta + \omega_x), \quad . \quad . \quad . \quad . \quad (4).$$

When the steam is released let the crank be in the position $O C_3$, then $\angle C_1 O C_3 = \omega_r$

First, suppose the valve to have *no* "inside lap." In this case the steam will be released when the valve is passing its middle position on its return stroke, *i.e.*, when the centre of the eccentric is at d.

$$\therefore \omega_r = \angle C_1 O C_3 = \angle E_1 O d = 180^\circ - \theta, \quad . \quad . \quad (5).$$

Next, suppose the valve to have "inside lap" $= L_i$. The release will then be delayed until the valve has moved past its middle position a distance $= L_i$. From O set off a distance, O M, along $O a = L_i$, and from M draw $M E_3 \perp O a$, meeting the eccentric circle in E_3. Then E_3 is the position of the centre of the eccentric when release takes place.

$$\text{Hence } \omega_r = \angle C_1 O C_3 = \angle E_1 O E_3 = \angle E_1 O d + \angle d O E_3 = 180^\circ - \theta + a,$$

$$\text{where } a = \angle d O E_3 \; \therefore a = \omega_r + \theta - 180^\circ, \quad . \quad . \quad (6).$$

$$\text{Now } \frac{O M}{O E_3} = \cos \angle E_3 O M = \sin a,$$

$$\therefore \frac{L_i}{\frac{1}{2} T} = \sin(\omega_r + \theta - 180^\circ), \text{ or } \frac{2 L_i}{T} = -\sin(\omega_r + \theta), \quad . \quad (7).$$

The "inside lead," or the amount the port is open for exhaust when the piston is at the end of its stroke, is shown on the valve diagram, p. 183, by the line U T. To find the position of the centre of the eccentric when the piston is at the end of its stroke. Produce $E_1 O$ to meet the eccentric circle again in E_4, then E_4 is the required position. From E_4 drop the perpendicular $E_4 K$ on the line $O a$. Then "inside lead"

$$= l_i = O K - O M = \tfrac{1}{2} T \sin \angle d O E_4 - L_i = \tfrac{1}{2} T \sin \theta - L_i$$

$$\therefore \frac{2(L_i + l_i)}{T} = \sin \theta, \quad . \quad . \quad . \quad . \quad (8).$$

From equations (2) and (8) we see that

$$L_o + l_o = L_i + l_i \quad . \quad . \quad . \quad (9).$$

which is also evident from the valve diagram, p. 183.

Next, let us find the position, $O\,C_5$, of the crank when compression begins. If the valve had *no* "inside lap," then compression would take place when the valve was in its middle position and moving in the opposite direction to that of the piston. Hence compression would take place on the return stroke when the centre of the eccentric was at c. It is, therefore, evident that the crank would then make an angle, θ, with the centre line of engine—*i.e.*, when there is *no* "inside lap"

$$\angle C_1\,O\,C_5 = \theta.$$

If, however, the valve has "inside lap," then compression will begin *before* the valve comes to its middle position. Produce $E_3\,M$ to meet the eccentric circle again in E_5, then E_5 is the position of the centre of the eccentric when compression begins.

$$\therefore \angle C_1\,O\,C_5 = \angle E_1\,O\,E_5 = \theta + \alpha,$$

$$\therefore \omega_c = 360^\circ - (\theta + \alpha), \quad . \quad . \quad . \quad . \quad (10).$$

Let us now find a general formula connecting together the "lap," "lead," "travel of valve," length of stroke of piston, and distance to point of cut off.

From C_2, the point of cut off, drop the perpendicular $C_2\,P$ on $C_1\,C_4$. Then (neglecting the obliquity of the connecting rod)—

$$C_1\,P = x, \quad O\,P = C_1\,P - O\,C_1 = x - \tfrac{1}{2}S,$$

$$\text{and } \frac{O\,P}{O\,C_2} = \cos \angle C_2\,O\,C_4 = \cos(180^\circ - \omega_x) = -\cos\omega_x.$$

$$\therefore \frac{x - \frac{1}{2}S}{\frac{1}{2}S} = -\cos\omega_x, \text{ or } \cos\omega_x = \frac{S - 2x}{S} \quad . \quad . \quad . \quad . \quad (11).$$

$$\text{But } \cos\omega_x = 1 - 2\sin^2\frac{\omega_x}{2}, \quad \therefore \sin^2\frac{\omega_x}{2} = \frac{1 - \cos\omega_x}{2} = \frac{x}{S}\,. \quad (12).$$

In a similar way, if y = distance from beginning of stroke to point of release, we can show that—

$$\sin^2\frac{\omega_r}{2} = \frac{y}{S} \quad . \quad . \quad . \quad . \quad (13).$$

From equation (1) we get—

$$\frac{\omega_x}{2} = 90^\circ - \theta + \frac{\lambda}{2}, \quad \therefore \cos\frac{\omega_x}{2} = \sin\left(\theta - \frac{\lambda}{2}\right),$$

and from equation (12), we get—

$$\frac{x}{S} = 1 - \cos^2\frac{\omega_x}{2} = 1 - \sin^2\left(\theta - \frac{\lambda}{2}\right).$$

Now, bisect $\angle E_1\,O\,E_3$ by the line $O\,E$, and from E draw $E\,H \perp C_1\,C_4$; then $\angle c\,O\,E = \theta - \dfrac{\lambda}{2}$.

Since $\dfrac{\lambda}{2}$ must be a very small angle, the perpendicular, $E\,H$, will almost exactly bisect $L\,N$. $\therefore O\,H = L_o + \tfrac{1}{2}l_o$.

$\sin\left(\theta - \frac{\lambda}{2}\right) = \sin \angle eOE = \cos \angle EOH = \frac{OH}{OE} = \frac{L_o + \frac{1}{2}l_o}{\frac{1}{2}T}$
$= \frac{2L_o + l_o}{T}$.

$$\therefore \frac{x}{S} = 1 - \sin^2\left(\theta - \frac{\lambda}{2}\right) = 1 - \left(\frac{2L_o + l_o}{T}\right)^2,$$

$$\therefore x = S\left\{1 - \left(\frac{2L_o + l_o}{T}\right)^2\right\} \quad . \quad . \quad (14).$$

We shall now apply the results we have arrived at to the solution of one or two examples To enable us to solve these examples we must have recourse to a table of natural sines, cosines, &c, which will be found in books on logarithms and most pocket books of formulæ, or at the end of this Text book. We may here collect together the results we have arrived at for the sake of reference

$$2\theta + \omega_a - \lambda = 180°, \quad . \quad . \quad . \quad . \quad . \quad . \quad (1).$$

$$\frac{2(L_o + l_o)}{T} = \sin\theta, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2).$$

$$\frac{2L_o}{T} = \sin(\theta - \lambda), \quad . \quad . \quad . \quad . \quad . \quad (3).$$

$$= \sin(\theta + \omega_x), \quad . \quad . \quad . \quad . \quad . \quad (4).$$

$$\omega_r = 180° - \theta \qquad \text{(no "inside lap")}, \quad (5).$$

$$= 180° - (\theta - a) \qquad \text{(with "inside lap")}, \quad (6).$$

$$\frac{2L_i}{T} = \sin a = -\sin(\theta + \omega_r), \quad . \quad . \quad . \quad (7).$$

$$\frac{2(L_i + l_i)}{T} = \sin\theta, \quad . \quad . \quad . \quad . \quad . \quad . \quad (8).$$

$$L_o + l_o = L_i + l_i, \quad . \quad . \quad . \quad . \quad . \quad . \quad (9).$$

$$\omega_c = 360° - (\theta + a), \quad . \quad . \quad . \quad . \quad (10)$$

$$\cos\omega_x = \frac{S - 2x}{S}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (11).$$

$$\sin\frac{\omega_x}{2} = \sqrt{\frac{x}{S}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (12).$$

$$\sin\frac{\omega_r}{2} = \sqrt{\frac{y}{S}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (13).$$

$$x = S\left\{1 - \left(\frac{2L_o + l_o}{T}\right)^2\right\}, \quad . \quad . \quad . \quad (14).$$

N.B.—In using the above formulæ it must be remembered that the obliquity of the connecting rod and eccentric-rod are not taken into account.

Example 1.—Given "travel of valve" = 5 ins., "outside lap" = $\frac{3}{4}$ in., "inside lap" = $\frac{1}{8}$ in, "angle of advance" = 20°. Find position of the crank (a) at admission, (b) at cut off, (c) at release, and (d) at compression.

Here $T = 5$ ins, $\quad L_o = \frac{3}{4}$ in, $\quad L_i = \frac{1}{8}$ in., $\quad \theta = 20°$.

(a) *To find position of crank at admission.*

From equation (3) we get—

$$\sin(\theta - \lambda) = \frac{2L_o}{T} = \frac{2 \times \frac{3}{4}}{5} = \cdot 3.$$

Referring to a table of natural sines we see that ·3 is the sine of an angle of 17° 28′,

$\therefore \sin(\theta - \lambda) = \sin 17° 28'$; or $\theta - \lambda = 17° 28'$, and $\lambda = 20° - 17° 28' = 2° 32'$, *i.e.*, the crank makes an angle of 2° 32′ with centre line of engine when steam is admitted.

(b) *To find position of crank at cut off.* From equation (1) we get

$$\omega_s = 180° - 2\,\theta + \lambda = 180° - 40° + 2° 32' = 142° 32'.$$

(c) *To find position of crank when steam is released.* From equation (7) we get

$$\sin\,(\,\theta + \omega_r\,) = -\frac{2\,L_i}{T} = -\frac{2 \times \frac{1}{3}}{5} = -\cdot 1333.$$

Now the angle whose sine is − ·13 must either be − 7° 40′ or 180° + 7° 40′ = 187° 40′. But $\theta + \omega_r$ cannot be negative, therefore we must take the other value.

$$\therefore\ \theta + \omega_r = 187° 40', \text{ and } \omega_r = 167° 40'.$$

(d) *To find position of crank when compression takes place.* From equation (10) we get

$$\omega_c = 360° - (\theta + a)\,;$$

and from equation (6) we get

$$a = \omega_r + \theta - 180° = 167° 40' + 20° - 180° = 7° 40'.$$

Substituting this in the last equation, we get

$$\omega_c = 360° - 20° - 7° 40' = 332° 20',$$

or compression takes place when the crank makes an angle of 27° 40′ with centre line of engine.

Example 2.—Given the "outside lap" = 1½ in., "inside lap" = ¼ in., "outside lead" = ⅛ in. Length of stroke of engine = 4 ft., cut off at ⅝ of stroke. Find (*a*) the "travel of the valve," (*b*) the "angle of advance," (*c*) "inside lead," (*d*) distance to point of release, (*e*) distance from beginning of stroke when compression begins.

Here $L_o = 1\frac{1}{2}$ in., $L_i = \frac{1}{4}$ in., $l_o = \frac{1}{8}$ in., $S = 4$ ft., $x = \frac{5}{8} \times 4 = 2\frac{1}{2}$ ft.

(a) *To find the "travel of the valve."* From equation (14) we get—

$$x = S\ \left\{1 - \left(\frac{2\,L_o + l_o}{T}\right)^2\right\},$$

$$\therefore\ 2\tfrac{1}{2} = 4\ \left\{1 - \left(\frac{2 \times 1\frac{1}{2} + \frac{1}{8}}{T}\right)^2\right\},$$

$$\therefore\ T = \frac{25}{\sqrt{24}} = 5{\cdot}1 \text{ ins.}$$

(b) *To find "angle of advance."* From equation (2) we get—

$$\sin\theta = \frac{2\,(L_o + l_o)}{T},$$

$$= \frac{2\,(1\frac{1}{2} + \frac{1}{8})}{5{\cdot}1},$$

$$= \cdot 6372,$$

$$\therefore\ \theta = 39° 35'$$

(c) *To find the "inside lead."* From equation (9) we get—

$$L_i + l_i = L_o + l_o,$$

$$\therefore\ l_i = L_o - L_i + l_o = 1\tfrac{1}{2}'' - \tfrac{1}{4}'' + \tfrac{1}{8}'' = 1\tfrac{3}{8}''.$$

(d) *To find distance to point of release from beginning of stroke.* From equation (13) we get—

$$\frac{y}{S} = \sin^2\frac{\omega_r}{2} = \frac{1 - \cos\omega_r}{2}.$$

Also, from equation (7) we have—

$$\sin(\theta + \omega_r) = -\frac{2\,L_i}{T} = -\frac{2 \times \frac{1}{4}}{5 \cdot 1} = -\cdot 098 = \sin 185^\circ\,38'.$$

$$\therefore\ \omega_r = 185^\circ\,38' - 39^\circ\,35' = 146^\circ\,3'.$$

$$\therefore\ \cos\omega_r = -\cos(180^\circ - 146^\circ\,3') = -\cos 34^\circ \text{ nearly} = -\cdot 829.$$

$$\therefore\ y = 4 \times \frac{1 + \cdot 829}{2} = 3 \cdot 66 \text{ ft.}$$

(e) *To find distance of piston from beginning of stroke when compression begins.* Referring to our figure (p. 186), draw $C_5 R \perp O C_1$. Then $C_1 R$ is the required distance.

Now, $C_1 R = O C_1 - O R = \frac{1}{2}S - \frac{1}{2}S \cos \angle C_1 O C_5 = \frac{1}{2}S\left\{1 - \cos(\theta + \alpha)\right\}$.

And $\alpha = \omega_r + \theta - 180^\circ$ . . . . . . equation (6).

$\therefore\ \theta + \alpha = \omega_r + 2\theta - 180^\circ = 146^\circ + 79^\circ\,10' - 180^\circ$, by (b) and (d),

$= 45^\circ\,10'$.

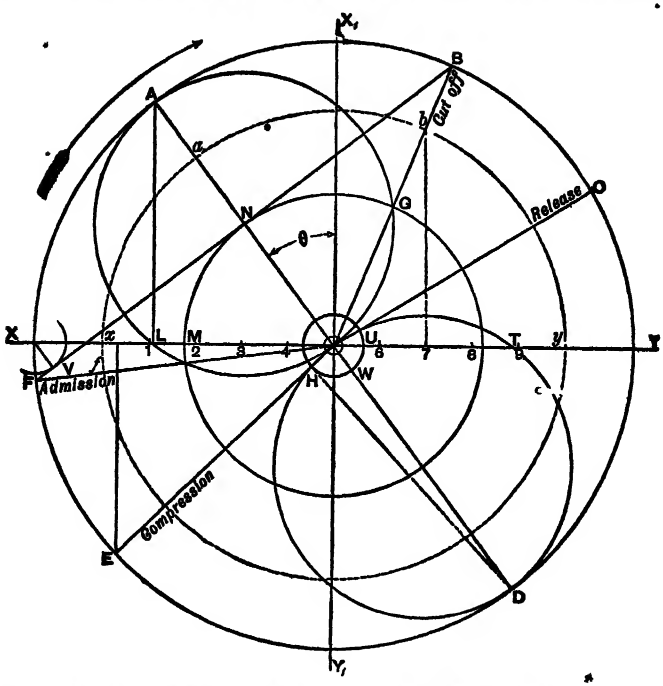
$$\therefore\ C_1 R = \tfrac{1}{2}S\left\{1 - \cos 45^\circ\,10'\right\} = 2\left\{1 - \cdot 705\right\} \text{ ft.} = \cdot 59 \text{ ft.}$$

Example 3.—In a steam engine the cut off takes place at ·7 of the stroke, the angle of lead is 6° 9′, the width of the steam port is $1\frac{1}{4}$ ins., and the steam port opens $\frac{17}{24}$ of its area. Find by Zeuner's diagram (neglecting obliquity) the travel of the slide, the angle of advance, the outside lap, and the outside lead. What would be the amount of inside lap necessary to produce cushioning after $\frac{6}{7}$ of the stroke has been performed? (S. and A. Hon. Exam., 1888)—cos 66° 25′ = ·4 ; cos 59° 52′ = ·502.

This question is evidently intended to be solved partly by calculation. In an examination tables are now allowed to be used, and therefore it will be interesting to solve, with the data given, how this question may be answered.

Draw the two axes, X O Y, $X_1 O Y_1$, at right angles to each other. With O as centre describe *any* circle $x\,b\,y$. Divide $x\,y$ into ten equal parts. From the seventh point of division, measured from x, draw $7\,b \perp x\,y$, meeting the circle in b. Join O b and produce it. The line O b represents the position of the crank when steam is cut off. From O draw O f below O X, making $\angle$ X O $f = \lambda = 6^\circ\,9'$ = angle of lead. Bisect $\angle f$ O b by the line O a. Then $\angle a$ O $X_1 = \theta$ = angle of advance. This is about all the length we can go with the construction in the meantime. We may, however, take our protractor and measure the angles θ and ω ($= \angle a$ O b).

We must now try and find either the outside lap or the travel of the valve. Since the width of the steam port is $1\frac{1}{2}$ ins.. and the maximum opening to



team is $\frac{17}{24}$ of the area of the port, then it follows that the maximum pening for steam $= \frac{17}{24} \times 1\frac{1}{2} = \frac{17}{16}$ ins Now, we have seen, p. 173, that—

$$T = 2\,(L_o + \text{max. opening for steam}),$$
$$= 2\,(L_o + k),$$

where k = max. opening for steam $= 1\frac{1}{16}$ ins.

$$\therefore \frac{2\,L_o}{T} = 1 - \frac{2\,k}{T}, \quad . \quad . \quad . \quad . \quad (a).$$

But $\frac{2\,L_o}{T} = \sin\,(\theta + \omega_x)$, equation (6), and $\theta = 90^\circ - \frac{\omega_x - \lambda}{2}$, equation (1).

$$\therefore \frac{2\,L_o}{T} = \sin\left(90^\circ + \frac{\omega_x + \lambda}{2}\right) = \cos\frac{\omega_x + \lambda}{2}.$$

Substituting this in (a), we get

$$\cos\frac{\omega_x + \lambda}{2} = 1 - \frac{2\,k}{T}, \quad . \quad . \quad . \quad . \quad (\beta).$$

Now we know that $\lambda = 6° 9'$, and we can now proceed to find ω_x. From equation (11), we have

$$\cos \omega_x = \frac{S - 2x}{S} = \frac{1 - 2 \times \cdot 7}{1} = - \cdot 4.$$

Hence from data given in the question, we see that

$$\omega_x = 180° - 66° 25' = 113° 35';$$

$$\therefore \cos \frac{\omega_x + \lambda}{2} = \cos \frac{113° 35' + 6° 9'}{2} = \cos 59° 52' = \cdot 502,$$

as given in question.

Substituting this in (β), we get

$$\cdot 502 = 1 - \frac{2k}{T} \therefore T = \frac{2k}{\cdot 498} = \frac{2 \times 1\frac{1}{16}}{\cdot 498} = 4 \cdot 26 \text{ ins.}$$

With centre O, and radius = 2·13 ins., describe circle X Y_1 Y, cutting Of, Ob, and Oa in F, B, and A. Join F B, cutting O A at right angles at N; then O N is the outside lap. With centre, X, describe an arc of a circle tangent to F B; then the radius, X Y, of this circle is the outside lead; or the outside lead may be got from M L. To find the amount of inside lap necessary to produce cushioning after $\frac{6}{7}$ of the stroke has been performed. Divide X Y into 7 equal parts, and from the first point of division next X, draw the perpendicular downward to meet the circle in E. Join O E. Then O E is the position of the crank when compression begins. Let E cut the valve circle, O D, in H. With centre, O, and radius, O H, describe the circle, H W U. The radius of this circle gives the inside lap. Since E H cuts the circle, O D, at a very small angle, it may be difficult to find the exact point, H, of intersection. This, however, can be easily overcome, remembering that the angle in a semicircle is a right angle. From D, draw D H $\perp$ E O, the point of intersection, H, is the required point.

LECTURE XIV.—QUESTIONS.

1. Sketch an eccentric and describe the several parts. What is the throw of an eccentric? Upon what does the amount of throw depend? What is the angle of advance?

2. What is the lap of a slide valve? Draw a section of a simple slide valve and ports, showing the valve (1) without lap, (2) with lap. For what purpose is "lap" given to a slide valve?

3. What effect is produced by putting lap on a slide valve? The lap on the steam side of a slide valve is $1\frac{1}{2}$ inches, that on the exhaust side is $\frac{1}{4}$ inch, and the lead is $\frac{1}{8}$ inch. Find the opening for exhaust which the valve gives at the lower port when the piston is at the top of its stroke. *Ans.* $1\frac{3}{8}$ inch.

4. Make a diagram showing a crank going backward, or opposite to the hands of a watch, and mark on the crank circle the points of admission, cut-off, release, and compression. Draw the probable curve of pressures underneath of a non-condensing engine showing the atmospheric line.

5. In a direct-acting horizontal engine the lengths of the crank and connecting-rod are 1 and 5 feet respectively. How far is the piston from the middle of its stroke when the crank is vertical? *Ans.* 1·23 inch.

6. Taking a direct-acting engine, and disregarding the effect of obliquity of the connecting rod, you are required to assign the proportion of lap to travel of slide valve, in order to cut off steam at $\frac{3}{4}$ of the stroke. *Ans.* $\frac{1}{4}$

7. Given that the travel of a slide valve is 5 inches, outside or steam lap $\frac{3}{4}$ inch, and the angle of advance $22\frac{1}{2}°$, find graphically the position of the crank at the point of cut-off. *Ans.* 140° from dead centre line.

8. In a direct-acting non-condensing engine let the crank be on the back dead centre. Sketch the slide valve and ports, marking the lap and lead. What is the object of putting inside lap to valve?

9. The stroke of the piston in a direct-acting engine is 4 feet, and the length of the connecting rod is 9 feet How far is the piston from the middle of its stroke when the crank has made $\frac{1}{4}$ of a revolution from a dead point? *Ans* 2·7 inches.

10. Travel of valve $= 8\frac{3}{4}$ ins.
Outside lap $= 2\frac{1}{4}$ „
Inside lap $= \frac{1}{4}$ „
Angle of advance $= 35°$.

Ans $\lambda = 4° 3'$. $\omega_x = 114° 3'$. $\omega_r = 148° 16'$. $\omega_c = 28° 16'$. $l_o = ·26$ in.

Find the points of admission, cut off, release, and compression, and the amount of lead by a Zeuner's diagram.

11. Given—
Outside lap $= 1\frac{1}{4}$ inch.
Maximum opening for steam $= 1\frac{1}{4}$ „
Cut-off at $\frac{7}{10}$ of the stroke.

Ans. T = 5 ins. $l_o = ·25$ „ $\theta = 37°$.

Determine the travel of valve, lead, and the angle of advance.

12. Given.
Travel of the valve $= 4\frac{1}{2}$ inches.
Outside lap $= 1$ „
Inside lap $= \frac{1}{4}$ „
Angle of advance $= 30°$

Ans. $3° 36\frac{3}{4}'$; $56° 23\frac{1}{4}'$; $23° 37\frac{1}{4}'$; $36° 22\frac{3}{4}'$. These angles are measured from the dead centres line.

Find the positions of the crank at admission, cut-off, release, and compression, also the lead of the valve.

13. A horizontal engine is constructed with a three-ported or locomotive slide valve and single eccentric for cutting off the steam at half-stroke. In what respects would you alter the working parts in order to cut off steam at three-quarters of a stroke? Explain by sketches the alterations which are necessary.

14. In a direct-acting engine, set out by a diagram the relative positions of the piston and crank during a stroke, on the supposition that the connecting-rod is of infinite length or remains parallel to itself. How is this diagram altered when a definite length is assigned to the connecting-rod?

15. The crank of an engine is 3 feet 6 inches, and the connecting-rod 9 feet long. Find the angle which the crank makes with the vertical when the piston is at half-stroke. *Ans.* 11° 12′ 44″.

16. Explain the effects produced by putting *outside* and *inside* lap respectively upon the slide valve of an engine. The outside lap is $1\frac{3}{4}$ inches, the lead is $\frac{1}{8}$ inch, and the greatest opening for steam is $1\frac{1}{2}$ inches, what is the travel of the valve, and how far is the valve from its middle position when the piston is just beginning its stroke? *Ans.* $6\frac{1}{2}$ ins.; $1\frac{7}{8}$ ins.

17. The length of crank is 14 inches, the slide-valve has half-travel of $2\frac{1}{2}$ inches, its lap is $1\frac{1}{2}$ inches, and its lead $\frac{1}{8}$ inch. At what distance from the end of the stroke will the piston be when the steam is cut off, if the angularity of the connecting-rod is neglected? Prove that the Zeuner diagram gives correct answers when the motions are simple harmonic. (S. & A., 1897, Adv.)

18. A link motion works a slide-valve; outside lap, 0·5 inch. By shifting the gear we get the following:—

Half-travel, . . .	2·50″	2·10″	1·70″	1·52″
Angle of advance, .	30°	40°	51°	69°

Find in each case the positions of the crank at *admission*, *cut-off*, *release*, and *compression*, and sketch the hypothetical indicator diagrams, taking any initial and back pressures you please. Neglect the angularity of the connecting-rod. (S. & A., 1897, Adv.)

19. Prove the correctness of the Zeuner valve diagram. A valve has an outside lap 1 inch, inside lap 0·3 inch. It is worked by a gear giving, in two positions, the following values of the half travel and advance:—

Half-travel, ...	3·12″	2·12″
Advance, ...	30°	51°

If the connecting-rod is five times the length of the crank, find the points of cut-off for both ends of the cylinder in both cases. What sort of gear might give the above conditions? (B. of E., 1900, Adv. and H., Part I.)

20. Steam is admitted to the cylinder of a double-acting engine at 80 lbs. per square inch. The back pressure is 17 lbs. per square inch. The friction of the engine may be taken to be represented by a back pressure of 8 lbs. per square inch on the piston. Find the cut-off to give maximum actual work per cubic foot of steam, taking "*pv* constant" as the law of expansion. Neglect clearance, cushioning, and condensation. If you use a formula for the average pressure, prove it correct. (B. of E., 1900, H., Part I.)

21. Show the position of a slide-valve at the beginning of the stroke of an engine. A slide-valve has half-travel 2·10 inches, advance 40°, lap

1 inch, inside lap 0·3 inch, draw a passible indicator diagram. Prove your valve diagram to be correct. (B. of E., 1901, Adv.)

22. The stroke of a slide valve is $3\frac{1}{4}$ inches, the outside lap is $\frac{3}{4}$ inch, and the inside lap is $\frac{1}{8}$ inch Find the maximum opening of the port to steam, the angular advance (the lead of the valve being $\frac{3}{16}$ inch), and the piston positions at cut-off, release, and when compression begins. You may neglect the effect of the obliquity of the connecting-rod. (C. & G., 1901, O., Sect. C.)

23. The travel of a D-valve is 3 inches, and the angular advance is 30°. Find the outside lap, so that the lead may be $\frac{1}{10}$ inch, and, neglecting the obliquity of the connecting-rod, find the position of the piston at the point of cut-off. Sketch the valve. (C. & G., 1902, O., Sec. C.)

24. A slide valve has a stroke of $4\frac{1}{2}$ inches, a lead of $\frac{1}{8}$ inch, and a lap of $1\frac{1}{8}$ inches: determine by construction at what point of the piston stroke the valve opens, shuts off steam, and opens release. Neglect effects due to obliquity of connecting-rod. (C. & G., 1902, O., Sec. C)

25. Sketch a simple slide-valve showing cylinder ports and no more of the cylinder; show the valve in its mid position. Show in dotted lines the position of the valve when the piston has just begun its stroke. What do we mean by outside lap of a valve, inside lap, advance, and half-travel? The half-travel is 3·36 inches, advance 42°. What simple diagram enables us to find the distance of the valve from its mid stroke for any position of the main crank? Prove it correct. Having such a diagram, we obtain the openings of the port to steam or exhaust by subtracting the outside or inside lap; explain how this occurs. (B. of E., 1903, Adv. and H., Part i.)

LECTURE XV.

CONTENTS.—Actual *versus* Ideal Conditions and Behaviour of Steam in the Cylinder of a Steam Engine—Loss of Pressure and Temperature, with Condensation, between Boiler and Engine Cylinder—Initial Condensation in the Cylinder—Devices for Reducing Cylinder Condensation—Steam Jacketing as a Preventive against Initial Condensation—Superheating as a Preventive against Initial Condensation—History of Superheated Steam—Prof. Ewing's 1899 Trials on the Schmidt System—Prof. Watkinson's Superheaters—Imaginary and Actual Steam Expansion Curves—The Real Benefits of Superheated Steam—Steam Separators—Effects of Clearance—Compression or Cushioning—Causes why Compression does not Return the Work Spent on it—Lead—Wire-drawing—Release—Compounding—Questions.

Actual *versus* the Ideal Conditions and Behaviour of Steam in the Cylinder of a Steam Engine.—In Lectures XII. and XIII., ideal, perfect, or imaginary conditions of the action of steam and of a gas doing work in a cylinder were considered. In Lecture XIV. the distribution of steam in a cylinder was dealt with, as effected and affected by the ordinary eccentric, slide valve, crank, connecting-rod and piston. Now, we have to consider:—

(1) In what condition ordinary steam from an ordinary boiler arrives at the engine stop valve. and how it is affected by its admission therefrom into the valve casing and the cylinder.

(2) How the steam behaves up to the point of cut-off, during expansion, release, exhaust and compression.

(3) How far the losses due to conduction, radiation, wire-drawing, clearance, and initial condensation are preventable, and what are the supposed as well as the actual benefits derived from high speeds, superheating, and from compound expansion in two or more cylinders.

Loss of Pressure and Temperature, with Condensation, between Boiler and Engine Cylinder.—Suppose that a boiler is capable of generating plenty of dry steam to keep an engine going continuously at its full normal load, then, if the steam pipes connecting the boiler and engine stop valves are short, straight, of sufficient size, and well-lagged with good non-conducting material, the drop in pressure and corresponding temperature between the boiler and the cylinder side of the stop valve, as

measured by two accurate steam gauges and thermometers, should not exceed 2 per cent. It, however, often happens, due either to too long or too small or imperfectly covered steam pipes, with perhaps several sharp bends, that the fall of pressure between a boiler and an engine stop valve exceeds 10 per cent., due to wire-drawing and friction. This mere drop in pressure, although it represents a direct loss in the potential energy of the steam, does not constitute the whole evil. For, undoubtedly, if the steam left the boiler as dry saturated steam, without any superheat in it, a portion would have liquefied on arrival at the valve casing. This very liquefaction bedews or wets the inner surfaces of the steam pipe and valve casing, the valve, ports, cylinder cover, and piston, thus causing still further liquefaction on account of the fact, that steam much more readily condenses upon a wet than upon a dry metallic surface. Further, wet steam leaks past valves and pistons much more easily than dry steam. Consequently, by the time that steam with only a 2 per cent. drop in pressure has passed from the stop valve into the cylinder through large, straight, short ports and begun to do its work upon the piston, it will have lost at least *other* 2 per cent. in pressure, and probably 10 per cent. of the weight of steam which left the boiler, up to the point of cut-off in each stroke, exists as water in the cylinder, even if the valve casing and cylinder are well lagged and covered with wood. Whereas, in the case of the steam which arrived at the stop valve with a 10 per cent. drop in pressure, if it has now to pass through long, narrow, bent ports, and if the cylinder and valve casing are not lagged, it may have lost more than *other* 10 per cent. in pressure and 30 to 50 per cent. of the weight of the steam which left the boiler dry, may now exist as water in the cylinder at the point of cut-off! Such a condition of affairs has been in the past of only too common occurrence, and it is perfectly evident, that such a mixture of steam and water cannot possibly expand in the cylinder, either isothermally or adiabatically. It is, therefore, not giving the good dry steam which left the boiler a fair chance to treat it in this crude, unscientific, and uneconomical manner.

Of late, far more attention has been given to this subject than was the case twenty-five to thirty years ago. The urgent demands due to severe local and international competition, for economical marine, factory, and electric plant engines, coupled with the simultaneous rapid advance and opportunities of scientific engineering education, have so combined to produce boilers, steam pipes and engines, wherein steam is generated, conducted, and worked, that now leave but little, to be still

further expected in the way of the economical production and application of steam as a motive power in the best up-to-date examples.

Initial Condensation in the Cylinder.*—For the purposes of argument and explanation, let us just suppose that the steam pipes are extra large and short, and that they have been well lagged with the very best commercial non-conducting material; further, that the valve casing and the cylinder are well lagged and covered with wood; also, that the steam ports are short and large, and that dry steam is admitted into the valve casing. Now, if the cylinder is one belonging to a simple condensing engine, its end, into which the steam is being admitted from the valve casing, must have been in direct communication with the condenser during the exhaust of the previous stroke. Consequently, this end must have been reduced to nearly the temperature of the exhaust steam, because the interchange of heat between the working skin surface of the cylinder and the exhausting steam is rapid. The temperature of the inside of an ordinary condenser may vary between 100° and 140° F., according to the vacuum and other circumstances, but we shall not be far wrong if we suppose, that the surfaces of the piston, cylinder cover, and port through which the steam just left this cylinder during the end of the previous stroke, had fallen to between 150° and 190° F Now, if the fresh, dry, entering steam be, say, 50 lbs. pressure absolute, its temperature will be 280° F.; and, naturally, the moment that it comes into contact with a conducting surface of only 150° to 190° F., it will at once part with a quantity of its heat with the result, that some of it instantly condenses upon the surfaces of the steam passages, cylinder cover, and piston, until these have been warmed up to a temperature somewhere between what they were and that of the entering steam. The very fact of these surfaces thus becoming wet induces still more steam to condense and the pressure to fall further than if they had been dry at the reduced temperature. *This liquefaction is termed "the initial condensation of the steam in the cylinder."* It accounts for a large proportion of the consumption of the steam in the cylinder of a simple condensing engine.

* Those who wish to study this subject thoroughly, should read the several papers by Bryan Donkin, Jr.; Profs. Dwelshauvers-Dery, Callendar and Nicolson, &c.; on "Cylinder Condensation, with Measurements of the Temperatures of the Cylinder Walls by Mercury and Platinum Thermometers," as found in the *Proc. Inst. C.E.*, vol. xcviii., pp. 250, 254; vols. c., p. [illegible]; cvi., p. 264; cxv., p. 263; cxx., p. 323; and cxxxi., p. 147. The [illegible] of these papers is the most important, because it refers to the previous papers, and to many careful thermo-electric measurements.

During the time that this steam is entering the cylinder up to the point of cut-off, part of its potential energy is being converted into work. If the pressure in the cylinder be kept constant up to cut-off, it shows that these initial losses are being replenished by fresh boiler steam and the drain on that steam by heat from the furnace.

When cut-off occurs and expansion of the steam really begins in the cylinder, still further liquefaction takes place and the pressure drops more quickly than it would otherwise have done, due to the mere transformation of heat into work, because fresh surfaces are being exposed by the piston. This liquefaction goes on until the temperature of the liquefied steam exceeds that of the steam in the cylinder. Then, re-evaporation takes place, and the pressure during the latter part of the stroke is kept up higher than it would be, due to the expansion of dry steam. Most of this re-evaporation, however, occurs during release and the exhaust stroke. This prevents the vacuum from being so good as it would otherwise have been, had the steam remained dry from admission to the end of the steam stroke. Consequently, a large portion of the heat of the entering steam, which is spent in raising the temperature of the cylinder, is uselessly thrown away in heating the condenser and creating an opposing back pressure.

Devices for Reducing Cylinder Condensation.—In addition to Watt's *separate condenser*, which is always used in the case of condensing engines, the following are the chief methods which have been tried for reducing the initial and subsequent condensation of steam in the cylinders of steam engines, although not necessarily in this precise order:—(1) Steam jacketing, (2) superheating, (3) steam separators, (4) reducing the clearance surfaces and volumes in the passages and cylinders, and (5) compounding. We shall endeavour to discuss these methods in this and in the subsequent lectures as fully as the space will permit, and give references to the more important papers which specially treat on them.

Steam Jacketing as a Preventive against Initial Condensation.—To prevent excessive initial condensation as well as the undue alternating give and take of heat between the working steam and the cylinder surfaces as far as possible, Watt invented a system which he termed "Steam Jacketing the Cylinder." This consists of passing steam direct from the boiler into an annular space surrounding the cylinder barrel, and sometimes into the cylinder covers, which are made hollow for that purpose. The idea was to keep the cylinder surfaces as hot as possible. It

however, really requires steam of considerably higher temperature than the initial temperature of the entering steam, with which to fill the jackets, before it can be of great use, and hence the hot air jacket devised by Mr. Edward Field.* Still, ordinary steam jacketing does materially help to prevent condensation inside the cylinder; and, as we shall see from the up-to-date practical examples which are illustrated and described further on, that this system is frequently adopted. It is simply a case of confining liquefaction of steam to the jacket, where it is harmless if the jacket is properly drained of both water and air, instead of permitting it to take place unduly inside the cylinder, where it is highly detrimental and wasteful; since, during the admission to and the expansion of steam inside the cylinder of a steam-jacketed engine, liquefaction must take place *in the jacket* in order to provide heat for the work being done inside the cylinder.

It is doubtful if Watt thoroughly understood the principle of the action of the steam jacket, and for a long time after he introduced it, engineers thought that it was simply a case of "robbing Peter to pay Paul." Of course, the surrounding of the working barrel and the covers of a cylinder with a steam jacket does not do away with the necessity for the lagging of the cast-iron outsides of the jacket with a good non-conductor. The better these are lagged, the more will conduction and radiation of heat be prevented to their outer surroundings; and, consequently, the more thoroughly will the heat in the steam jacket penetrate inwards to the working steam. Besides, it ensures a cooler, and hence a more comfortable, engine-room if as little heat as possible passes into it.

Economy Due to Steam Jacketing.—(1) This increases the earlier the cut-off. (2) It increases the less the diameter of the cylinder. (3) It decreases as the piston speed and revolutions increase. (4) It decreases the higher the entering steam is superheated. For example, it is of little use to jacket the H.P. cylinder of a compound engine when the supply steam is superheated by 200° F. or more.

* See the Author's *Text-Book on Applied Mechanics and Mechanical Engineering*, vol. i., Lecture VIII., for a description, with his tests, of "Field's Combined Steam and Hot Air Engine." Field forced hot air obtained from pipes situated in the boiler flues into the jacket of his engine at an entering temperature of over 574° F., whilst the initial temperature of his steam was only 345° F. There was, consequently, a continual transfer of heat from this higher temperature air through the walls of his cylinder, which effectually prevented the slightest condensation of the working steam. With this *non-condensing single-cylinder* engine, the steam used was only 18·6 lbs. per I.H.P.-hour.

Superheating as a Preventive against Initial Condensation.*—In Lecture VIII. we explained what was meant by superheating steam. We showed there, that if heat be added to dry steam its temperature is raised without increasing its pressure, and the more it is heated the nearer does it approach to the physical qualities of a perfect gas. Consequently, if a convenient and economical means be adopted of imparting the necessary heat-units to steam, which shall ensure not only its arrival at the cylinder in a perfectly dry state, but also prevent initial condensation during the time of its admission and during expansion, the whole of the troubles of leakage past pistons and valves, as well as the condensation (which we have been discussing), would at first sight appear to be overcome. In order to explain how this desirable object has not been so easily accomplished, and how it happened that its adoption has recently been revived and strongly advocated, we shall devote a few pages to the history of this subject, to the means by which it is produced, to the precautions which must be observed in using it, and to some of the current misconceptions regarding its thermo-dynamic properties and its capabilities.

History of Superheated Steam.†—From an early date in the use of steam as a motive power, it was recognised, that the steam generated in ordinary steam boilers contained more or less water in suspension. It was also known, that if this suspended water—or *priming water*, as it is usually called—was converted into *dry steam*, or still better, into *superheated steam*, economy would result in its use in the steam engine. The precise physical qualities, functions, and most reliable methods of creating as well as of using highly superheated steam are still engaging the special attention of engineers.

Although John Payne undoubtedly produced superheated steam in his "flash" boilers in 1736, Sir William Congreve treated steam after its formation in a boiler in 1821, and Jacob Perkins produced such steam in 1822, it was not until 1832 that Trevithick patented and seemed to understand the economy derivable from the use of superheated steam. We are, however, indebted to the investigations of Hirn for the first really useful trials and scientific explanations of its properties, as found in the

* For the formula, calculations, and curves showing the percentage gain due to the use of superheated steam with different engines, see Index for "Gain with Superheated Steam."

† The student who desires to study this subject fully should read a paper on "Superheated Steam," by F. J. Rowan, A.M.Inst.C.E., and the discussion thereon, as found in vol. xlvii., *Trans. Inst. Engs. and Ships. in Scotland* (Glasgow, 1903-4), where other papers and sources relating to this interesting and instructive subject are mentioned.

Bulletin of the Industrial Society of Mulhouse and Alsation Society of Steam Users. He patented a form of superheater in 1855 which he called the "Hypo-thermo-generator."

About the same time (1854 to 1864), Mr. B. F. Isherwood, Chief Engineer, U.S. Navy, carried out a series of experimental researches to ascertain the comparative economy of steam with different measures of expansion, the causes and quantities of the condensations in the cylinder, the economic effect of steam jacketing and steam superheating, &c.*

Mr. John Penn, the well-known marine engineer of Greenwich, fitted superheating apparatus into the S.S. "Valetta" in 1857, and thus saved 20 per cent. in the consumption of fuel. Many other eminent engineers followed his example, such as Boulton & Watt, with Scott Russell, for the "Great Eastern" in 1862, thus superheaters for marine engines became more or less popular during these 10 or 12 years.†

The author remembers bearing a hand in pulling out some of these installations whilst serving his apprenticeship at the Aberdeen Iron Works in 1868, because considerable trouble had been experienced from their pitting and corrosion, as well as from the difficulty of lubricating the pistons, slide valves and their rods. Although the pressure of steam did not exceed 20 lbs. by gauge at that time, yet the temperature of the superheated steam was such, that it either decomposed or volatilised the ordinary fatty and vegetable oils then used for lubrication purposes. Seeing that Randolph, of Randolph & Elder, Glasgow, had successfully made and applied compound marine engines by 1868, with better commercial results and with far less trouble than any known combination of superheaters and simple condensing marine engines, superheating fell into disfavour until they were revived about ten years ago, in 1895.‡

Prof. Ewing's 1899 Trials of the Schmidt System.—In October, 1899, Prof. J. A. Ewing, F.R.S., of Cambridge University, tested "The Schmidt System" at Sheffield on a single-acting engine with two horizontal cylinders lying side by side. This

* See *Experimental Researches in Steam Engineering*, by Chief-Engineer B. F. Isherwood, U.S. Navy, published by Wm. Hamilton, Hall of the Franklin Institute, 1863.

† See *Proc. Inst. C.E.*, 1860; *Trans. Inst. M.E.*, 1859-60, and "Bourne on the Steam Engine."

‡ See *Trans. Inst. Mech. Engs.*, April, 1896, for a paper by Wm. Patchell, *et seq.*; and Prof. W. C. Unwin's paper on "The Determination of the Dryness of Steam." *Proc. Inst. C.E.*, vol. cxxviii., May, 1897, for Prof. Wm. Ripper's paper. Also, *Trans. Amer. Soc. Mech. Engs.*, vol. xvii., May, 1896, for Prof. R. H. Thurston's paper on "Superheated Steam."

engine had pistons of long trunks without piston-rods or stuffing-boxes, and cranks set at 180° apart. The pistons were 7·09 inches diameter, stroke 11·8 inches, and made 175 revolutions per minute. Steam was generated in a vertical boiler with cross tubes, and above the vertical flue was fixed the superheater. When the boiler pressure was 129 lbs. per square inch by gauge, the temperature of the steam leaving the superheater was 387° C. and 338° C. at the engine. This small condensing engine of 20·1 I.H.P. only used 17·7 lbs. of steam per I.H.P.-hour, as measured by weighing the condensed steam. This excellent result, as well as others carried out by the same person on other plants of the same make, gave a special impetus to inventors and steam engine users, more especially to those in charge of central electrical stations. Since then, many other patented forms have been tried more or less successfully and economically. With the use of the high flash-point oils and the adoption of valves and superheater tubes made of special metals to resist erosion and corrosion, they have overcome the difficulties of lubrication and chemical action so freely experienced by the previous generation of engineers.

Prof. Watkinson's Superheaters.*—The present Professor of Steam Engines and other Prime Movers at the Glasgow Technical College is one of the most successful advocates of steam superheating. Early during this last wave of resuscitation in its favour, he devised two types of superheaters on the *independently fired* and the *shunt circuit* systems. Fig. 1 is a reduced illustration of a photograph taken during the construction of an independently fired superheater in the Scotstoun Iron Works of Messrs. Mechan & Sons, Glasgow, and which has been working for some years in a Lanarkshire Colliery.

FIG. 1.—PROF. WATKINSON'S SUPERHEATER FOR INDEPENDENT FIRING.

The three "headers" or straight pipes seen at the bottom of the photograph are usually made of mild steel of 10 to 12, or

* See *Proc. Inst. N.A.*, June, 1903, for his paper on "Some New Types of Superheaters." I am indebted to this Institution and to Prof. Watkinson for permission to reproduce the three figures with my own system of index lettering, and make this reference to his paper.

more, inches in diameter, according to the size of the installation, whilst the inverted U tubes are $1\frac{1}{4}$ to $1\frac{1}{2}$ inches of solid drawn steel. These tubes are so arranged in rows and at such small distances apart, that the products of combustion from the furnace must pass between them in thin divided sheets. The object of causing the heated gases to be split up in this way is, that they may the more readily part with their heat to the tubes. Ordinary steam (wet or dry) from the boiler stop valve, first passes into the "inlet" header, then through one set of tubes to the "intermediate" header, from it through the second set of tubes to the "outlet" header, and from it to the stop valve at the engine in a more or less superheated condition, by regulating the flue dampers. The products of combustion flow transversely through between the tubes, where the temperature of the approaching gases may be white hot, whilst they may leave them at 450° F. or thereby. Consequently, care must be taken, that neither the tubes suffer, nor the steam be over-superheated, nor the gases leave at too high or too low a temperature. One thing is clear, that the tubes in this superheater have great freedom of expansion and contraction, due to their shape and to their simple attachment to the headers; for, they are merely passed through drilled holes in the latter and fixed thereto by a boiler tube expander, applied from the insides of the headers in the ordinary way.

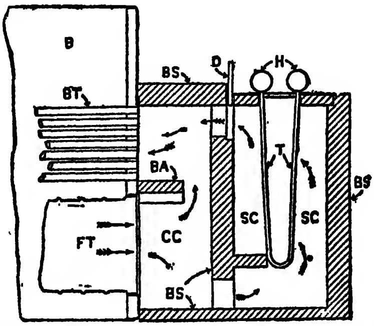


FIG. 2.—PROF. WATKINSON'S SUPERHEATER FOR DRYBACK LAND BOILERS.

Fig. 2 shows the application of a *shunt circuit* superheater to an ordinary "dryback" land boiler. Here the furnace gases pass, as indicated by the arrows, from the furnace flue tube F T, of the boiler B, to the combustion chamber C C, and strike against the inside of the brick setting B S. Then, according to the regulation of the damper door D, more or less of the gases

pass upwards and around the end of the baffle arch B A, and through the boiler tubes B T. The remainder of the gases pass on to the superheater chamber, S C, through between and around the superheater U tubes, T (as shown hanging freely from their two headers, H), to the opening left by the elevated damper, D, the boiler tubes as before, and from thence to front smoke box and up the chimney. When the damper, D, is closed, none of the gases pass through the superheater.

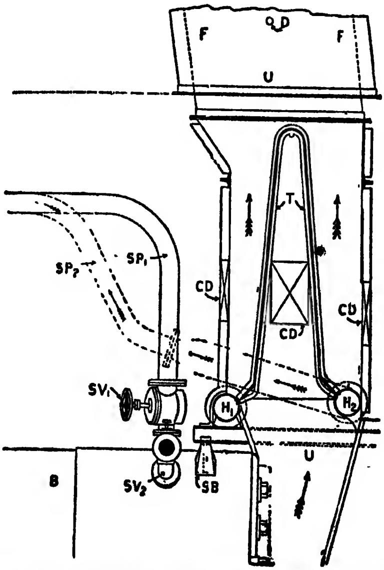


FIG. 3.—PROF. WATKINSON'S SUPERHEATER FOR MARINE BOILERS.

Fig. 3 shows this superheater placed above the front smoke box of a marine boiler B, inside the uptake U, below the funnel F. Here the gases which leave the boiler tubes for the uptake U, pass by the two headers H_1, H_2, through between the inverted U superheater tubes T, to the funnel F, where the throttle valve type of damper D, is indicated by an end view of its spindle. Ample provision is shown by four cleaning doors C D, for getting at the superheater tubes. The headers H_1, H_2, rest upon wrought-iron beams upheld by supporting brackets S B, fixed to the boiler shell.

When ordinary steam is desired, the main stop valve S V_1, is opened and the steam passes direct from the boiler B, through

the steam pipe, $S P_1$, to the engines. But, when superheated steam is desired, $S V_1$ is closed and $S V_2$ is opened, which permits boiler steam to flow into header H_1, and the dried superheated steam is then taken from header H_2, by the steam pipe $S P_2$, to the engines.*

Imaginary and Actual Steam Expansion Curves.—Having discussed how ordinary wet, dry, and superheated steam arrives from boilers and superheaters at the engine stop valve, and how liquefaction takes place in the cylinder of an ordinary simple condensing engine, it will now be advisable to distinguish between the ideal or imaginary curves of gas, steam, and air, and that of the actual expansion curves derived from ordinary and from superheated steam. We shall deal, later on in this and in future lectures, with the other devices which have been tried for preventing liquefaction in a cylinder, and show how indicator diagrams are taken and measured.

Curves 1 and 2 are Isothermal Expansion or Hyperbolic Curves, where $P V = $ *Constant.*†—Referring to Fig. 4, imagine (as we did in the previous lectures), that a perfect gas is admitted at 120 lbs. absolute per square inch to a non-conducting cylinder (having no clearance space between its piston and the cylinder cover), then this pressure may be depicted by the ordinate line, O A. Let the gas begin to press forward the piston slowly for $\frac{1}{10}$ of the full stroke, as represented by the line, A C. If the gas be now cut off sharply from the cylinder whilst its temperature is (somehow or other) maintained constant, it will expand and force the piston forward to the end of its stroke. If this expansion be performed *very slowly* (so that the gas may still be kept at constant temperature), the pressure of the gas will so fall, that ordinates drawn up from the horizontal line of volumes, O B, at any number of points along the stroke, will represent (to the same scale as O A), the pressures which would exist in the

* This superheater has been fitted not only to German Ocean steamers, but to Trans-Atlantic liners, and the results are being now watched and noted, not only by the professor, the maker, and the owners of these ships, but also by the large army of naval and mercantile marine engineers. Unfortunately, as far as this edition is concerned, reliable, definite data *re* the results of these trials have not yet been obtained over a sufficiently long period to place them before my readers. But, in some experiments carried out by Prof. Watkinson with his "shunt circuit" type connected to a Lancashire boiler of 8 feet diameter and 30 feet long, he says that he found a saving of 27·4 per cent. in coal over that with the same boiler without his superheater.

† The student should here refer to *The Engineer*, May 29th, 1903, p. 535, for Prof. Robert H. Smith's article on "The Expansion, Separation, and Compression of Wet Steam." Also, to *The Electrical Review*, May 27th, 1904, p. 866, for Mr. W. H. Booth's article on "Steam Curves."

cylinder behind the piston, at each point along the stroke. The curved line No. 2, drawn through the upper ends of these pressure ordinates will therefore depict graphically the expansion of the gas within the cylinder, and the equation to this curve is that for the hyperbola, viz.:—

$$PV = \text{constant.} \qquad \text{Or, } P \propto \frac{1}{V}.$$

Where P = pressure per square foot, and V = volume in cubic feet.

Because, the pressure of an enclosed perfect gas which is kept at constant temperature varies *inversely* as its volume. In the same way, if we abide by the foregoing conditions and let the cut-off be at D instead of at C, the curve No. 1, or D E, will then represent the isothermal expansion of the gas.

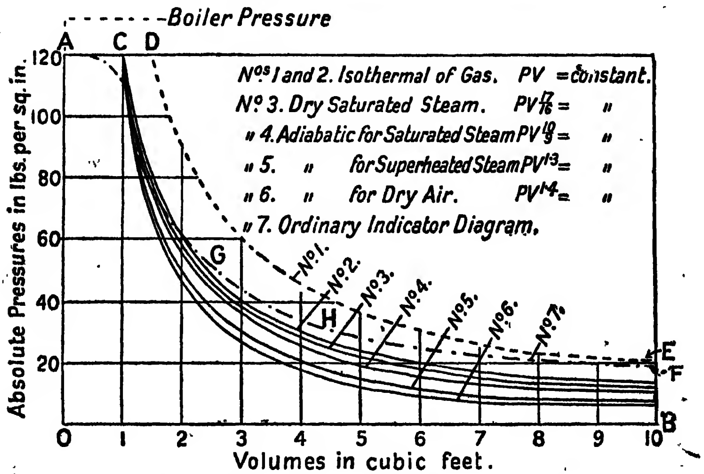


FIG. 4.—GAS, STEAM, AND AIR EXPANSION CURVES.

It is evident, that the special restrictions and imaginary conditions here assumed, do not pertain to those of any real cylinder made of metal or to any known means of keeping steam *at constant temperature* whilst expanding, and at the same time doing work. Consequently, these two curves, Nos. 1 and 2, do not represent the actual conditions under which ordinary steam expands in a cylinder. They are never obtained in practice, although they are convenient for showing the limit to which a perfect gas may attain whilst expanding isothermally.

Curve 3—Dry Saturated Steam Curve, where $PV^{\frac{17}{16}}$ = *Constant.*—If dry saturated steam be admitted at 120 lbs. per square inch absolute, into a well-jacketed cylinder for $\frac{1}{10}$ of the stroke (as represented by the line, A C, in Fig. 4) and then allowed to expand, it would fall in pressure as represented by Curve No. 3, if, during the whole time of the said expansion, the steam just received sufficient heat from the jacket to keep it dry. Prof. W. J. M. Rankine devised the following equation to represent this curve :—

$$P \cdot V^{\frac{17}{16}} = P V^{1\cdot} \quad \text{constant, } C = 69{,}000 \text{ foot-lbs.}$$

The student should here refer to the figure in Lecture XXVI. on the De Laval turbine, which has been accurately drawn, representing this curve when a volume of 1 cubic foot is plotted horizontally to the same scale as 1 lb. pressure vertically. In fact, it is only by so drawing these curves to a uniform scale that the eye can readily detect their variations in form and slope.

This curve of expansion may be closely followed by admitting superheated steam to a cylinder, so that the extra heat units of superheat are just expended in doing work, and in heating the cylinder up to the point of cut-off. Thereafter the steam must just get sufficient heat from a jacket to keep it dry to the end of the stroke. As will be seen from the practical examples given later on, engineers do frequently aim at producing this high-class condition of affairs, and they consequently compare the efficiency of their results by drawing this curve on their combined indicator diagrams.

Curve 4—Adiabatic Expansion of Saturated Steam, where $PV^{\frac{10}{9}}$ = *Constant.*—If a gas neither receives nor rejects heat as it expands or is compressed, then the curve which gives the relation between its pressure and its volume at each instant, is termed an adiabatic curve. Hence the work done by a gas when expanding adiabatically is all performed by a proportional loss of own internal initial heat energy. When a gas is being compressed adiabatically, then the whole of the work spent upon compressing it goes to increase its internal energy. Consequently, adiabatic expansion of steam could only be perfectly realised, if it were expanded or compressed without any change in its nature, in a perfectly non-conducting, non-radiating cylinder. This condition of affairs is never actually realised with steam, because, whilst it expands and does work, it loses heat in proportion to the work done and to conduction between it and the metal cylinder. But, the greater the piston speed and number of revolutions per minute, the less is the time and

opportunity for this latter effect to take place. Hence, we gain so far in economy by high piston speed and an increase in the revolutions per minute.

Although this curve is not strictly followed by the expansion of steam, yet (as will be seen later on) engineers do frequently compare their performances by plotting it over their combined indicator diagrams. • In the case of the De Laval and some other steam turbines, this curve does actually express the rate by which the steam pressure diminishes as its volume increases. This is clearly shown by the curve which has been carefully drawn to the same scale for 1 lb. pressure and 1 cubic foot volume, just referred to in Lecture XXVI.

$$\text{Here,} \qquad P V^{\frac{10}{9}} = P V^{1\cdot1}. \quad \text{Or, } P \propto \frac{1}{V^{1\cdot1}}.$$

Referring again to Fig. 4. If dry saturated steam were admitted into a perfectly non-conducting, heat-opaque cylinder at 120 lbs per square inch absolute for $\frac{1}{10}$ of the stroke, and allowed to expand therein without receiving or gaining heat from any source whatever, it would follow Curve No. 4, which falls below the previous curves.

Curve 5—Adiabatic Expansion of Superheated Steam, where $P V^{1\cdot3}$ = *Constant.*—If superheated steam were admitted to a cylinder at 120 lbs. pressure per square inch absolute and cut off at $\frac{1}{10}$ stroke, as represented by A C in Fig. 4, then, if this steam remained in a superheated condition up to the end of the stroke, it would expand according to the formula :—

$$P V^{1\cdot3} = \text{constant.} \quad \text{Or, } P \propto \frac{1}{V^{1\cdot3}}.$$

However, this would entail in an ordinary steam cylinder doing ordinary work a very high initial degree of superheat, which, as will be explained later on, is inadvisable unless certain precautions are observed.

Curve 6—Adiabatic Expansion of Dry Air, where $P V^{1\cdot4}$ = *Constant.**—Following the same reasoning in the case of dry air, we see, that the curve plotted to the equation,

$$P V^{1\cdot4} = \text{constant.} \quad \text{Or, } P \propto \frac{1}{V^{1\cdot4}},$$

yields Curve 6 in Fig. 4. This is the lowest of all the curves,

* Prof. Rankine in his text-book on *The Steam Engine* gives the power for dry air as 1·408; Perry, 1·414; and Ewing, 1·404 as well as 1·408; but for our purposes here, 1·4 will be sufficiently accurate.

and hence the least amount of work will be done by air per cubic foot when expanded adiabatically.

The following table shows the *mean pressure as a percentage* of the initial pressure or work obtainable by each of the foregoing expansion curves when cut off at $\frac{1}{10}$ of the stroke in a cylinder without clearance :—

PERCENTAGE RATIO OF MEAN TO INITIAL PRESSURE WITH 10 EXPANSIONS.

Isothermal. $PV = \text{Const.}$	Dry Saturated Steam. $PV^{\frac{17}{16}} = \text{Const.}$	Adiabatic Saturated Steam. $PV^{\frac{10}{9}} = \text{Const.}$	Adiabatic Superheated Steam. $PV^{1.3} = \text{Const.}$	Adiabatic Dry Air. $PV^{1.4} = \text{Const.}$
33 °/₀	31·4 °/₀	30·3 °/₀	26·6 °/₀	24·5 °/₀

Curve 7—From an Indicator Diagram.—If dry saturated steam of the boiler pressure shown in Fig. 4 had arrived at the cylinder with a pressure of 120 lbs. per square inch absolute, and were cut off at $\frac{1}{10}$ stroke, its pressure would, under ordinary circumstances, follow the chain dotted line, A G H F. Here, we see, that the pressure has dropped about 10 per cent. between the boiler and the cylinder, and that the initial pressure falls during the time of admission, due to wire-drawing and initial condensation.

When cut-off takes place, the pressure drops quickly, due to further cylinder condensation, for about another $\frac{1}{10}$ of the stroke; whereas, thereafter, re-evaporation of the previously condensed steam takes place and the pressure keeps up higher than it would have done had there been no initial condensation. Here, the points G, H, and F, represent the beginning, intermediate, and final pressures due this re-evaporation. In reality, if we assumed that there was 30 per cent. of initial condensation at the point of cut-off C, then the weight of steam admitted to the cylinder would have occupied a volume represented by the distance A D, before expansion began. We thus see, that there is a considerable area of lost work from the boiler pressure line down to about the point G, and that we never realise anything like the theoretical full area which the steam could give out, as represented by the figure, O A D E B, if it were supplied as a perfect gas at the given pressure and expanded according to the imaginary isothermal curve, D E.

The Real Benefits of Superheated Steam.—From an examination of these seven curves and the percentage table of mean pressures obtained with ten expansions, it is evident that super-

heated steam does not give such a good return in work done *per unit volume* as dry saturated steam. Under the conditions noted in that table, we observe that the mean pressure is only 26·6 per cent. for superheated steam, as against 31·4 per cent. for dry saturated steam, thus showing that the latter, if kept dry to the end of the stroke, yields fully 15 per cent. more work for the same volume and cut-off. Consequently, we conclude, that from a mere thermodynamic point of view, superheated steam does not yield such a good return as dry saturated steam. This arises from its low specific heat. Of course, per unit weight, superheated steam would contain more heat units than dry saturated steam and give out more work. The real benefits of superheated steam, as we have just seen, come into play in its prevention of initial condensation in the cylinder and in the paradoxical difficulty which it experiences in leaking past moving valves and pistons. We shall frequently refer to the applications of superheated steam later on in these lectures, and shall then take the opportunity of stating how it is generated, used, and compared with ordinary steam under different circumstances.

Steam Separators.—In order to prevent, as far as possible, the introduction of wet or "priming" steam from a boiler entering the valve casing of an engine, a simple device, termed a steam separator, is often interposed between the steam pipe and the engine. As will be seen by referring to the compound Bellis-Morcom engine, this separator consists of a cast-iron chamber into which the wet steam flows and impinges upon a diaphragm plate, thus causing the heavier condensed particles to fall to the bottom of this chamber, where they may be drawn off to the condenser, whilst the separated steam flows upwards to the engine stop valve and valve casing. Of course, all the suspended moisture cannot be trapped in this way, but, nevertheless, the steam is considerably dried and the working of the engine improved.

Effects of Clearance.—In actual practice, the piston does not come close up to the end of the cylinder at the end of its stroke, a small space being of necessity left between the piston and the cover to allow for the wear of the journals and other causes. Besides this, there is the volume of the steam ports between the valve face and the cylinder. This combined space between the piston and the cylinder cover, *plus* the steam ports, is termed *the clearance* of the cylinder, and exercises an important influence upon the expansion of the steam; for it must be filled with steam at the moment of cut-off, and the volume of steam expanding is equal to the volume of the cylinder to the point of cut-off + the space at the end of the cylinder + the volume of the steam ports.

The ratio of expansion of steam in a cylinder, as usually understood, is

$$= \frac{\text{the vol. of cylinder}}{\text{vol. to point of cut-off}}, \text{ or, } \frac{\text{area} \times \text{length of stroke}}{\text{area} \times \text{distance to pt. of cut-off}};$$

but, if clearance be taken into account, the *true* ratio of expansion is much less than the ratio given above.

Let c = fraction of cylinder's capacity representing clearance,
,, r = ratio of expansion *as above*.
,, r_1 = true ratio of expansion.

$$\text{Then, } r_1 = \frac{\text{vol. of cylinder} + \text{clearance}}{\text{vol. to pt. of cut-off} + \text{clearance}} = \frac{1+c}{\frac{1}{r}+c} = \frac{r(1+c)}{1+cr}.$$

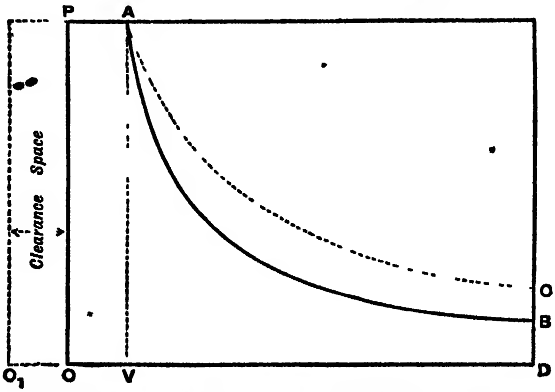


FIG. 5.—EFFECT OF CLEARANCE ON EXPANSION CURVES.

The difference between r and r_1 is obviously greatest with high ratios of expansion, or early cut-off, when $\frac{1}{r}$ may often be equal to c; hence, with high ratios of expansion the clearance space should be reduced to a minimum. An example may make this clearer. Let steam be cut off at $\frac{1}{8}$ of the stroke, then $r = 8$. Also, let the clearance capacity $= \frac{1}{8}$ of the capacity of the cylinder.

$$\text{Then,} \qquad r_1 = \frac{r(1+c)}{1+cr} = \frac{8(1+\frac{1}{8})}{1+1} = \frac{9}{2} = 4\tfrac{1}{2}.$$

Therefore, the volume of steam admitted to the cylinder is really only expanded $4\frac{1}{2}$ times instead of 8 times; and it is easy to see that the curve of expansion will be materially affected thereby. Fig. 5 shows curves of expansion from P V = constant.

A B is the curve which would be followed by the steam expanding 8 times with a volume, O V ($\frac{1}{8}$ O D), and pressure, O P. A C is the curve of expansion, which is really followed by the steam when the clearance space is taken into account, the volume being O_1 V, the pressure the same as before, and the expansion then being only $4\frac{1}{2}$ times. This shows the importance of taking the clearance into account in considering indicator diagrams.

In practice it is impossible to avoid clearance altogether, but the losses arising from it may be considerably reduced by closing the exhaust before the end of the return stroke.

Compression or Cushioning.—This is effected, as we pointed out in Lecture XIV., by closing the exhaust port before the piston has completed its return stroke. Then, any steam still remaining in the cylinder is compressed into the clearance spaces. If the compression were so great as to raise the pressure of the steam in the clearance spaces up to the initial pressure of the steam, loss from the clearance spaces would be largely avoided, since these spaces would already be full of steam at the initial pressure, when the piston began its next steam or forward stroke.

Causes why Compression does not Return the Whole of the Work Spent on it.—The total return for work spent on compression could only be strictly true of an engine which expanded right down to the back pressure line before commencing to exhaust. Or, in the case of an engine whose indicator card had only *four* sides; in fact, a *Carnot's reversible engine*, where the cooling takes place entirely by expansion and the heating entirely by compression. The indicator card of an ordinary steam engine has *five* sides, and, as there is a sudden drop of temperature from the point of release to that of the exhaust back pressure, we actually expend more work upon compressing the clearance steam during a return stroke than it usefully exerts on the next steam or forward power stroke.

In addition to this, we know, that the exhaust steam at the point where compression commences is, as a rule, dry steam. If so, the compression will take place adiabatically. But, during the forward stroke, this compressed steam mixes with fresh steam from the boiler, and, as a rule, initial condensation has then taken place. *Now, when steam expands in presence of water it does not and cannot expand adiabatically.* Hence, from this cause also, we see, that the previously adiabatically cushioned steam does *not exert* during expansion the same energy as was put into it during compression. Add to these two circumstances the fact, that heat is always radiating from a cylinder and we see, that compression during exhaust can never

yield up the full work done upon it. Hence, the less the volume of the clearance spaces the better, as in the most efficient Corliss engines.

The mean pressure of steam would, however, be greatly reduced by such excessive cushioning. The useful extent of cushioning, considered with reference to the motion of the engine alone, depends chiefly on the speed of the engine. In very fast-running engines a large amount of cushioning is necessary, in order to check the momentum of the moving parts gradually, and reverse the direction of motion without shocks; but, if the piston speed be slow, a less compression will suffice to keep the motion smooth and free from jerks (see Lecture XVIII.). These considerations limit the amount of compression to be used for any particular case. In engines having a high ratio of expansion and great piston velocity, the exhaust steam might with advantage be compressed up to the initial pressure, but, in other cases, a moderate compression is all that can be recommended. The effect of compression on the indicator diagram is a sudden rise in the exhaust or back pressure line just before steam enters, and is shown on the following diagram by the line, *m n*.

Compression up to the initial pressure of the steam has a further advantage in unjacketed cylinders, viz., that the cylinder becomes heated up to the initial temperature of the steam by the work done upon it, and condensation of the entering steam may, therefore, be greatly reduced.

Lead.—It is necessary in practice, especially with high-piston speeds and low-pressure steam, to open the slide valve before the piston has reached the end of its stroke, in order to assist the compression and maintain the full initial pressure as the piston moves forward. This is shown by the black heavy line, *a* to *cut-off*. This amount of opening is termed the "lead" of the valve. If no lead be allowed, the valve is not sufficiently open when the piston begins to move forward, and the full pressure of steam does not come upon the piston until it has travelled over a part of the stroke. The loss is shown by the sloping-down corner, *a b*, in Fig. 6.

Wire-Drawing.—When the steam comes from the boiler through too small steam pipes, or through small superheater pipes, or when it enters the cylinder through contracted ports, or is prevented from following up the piston at full pressure, it is said to be *wire-drawn*. The effect upon the indicator diagram is a fall of pressure shown by the dotted line, *a b d*. With a common slide valve, a certain amount of wire-drawing will always take place at the point of cut-off, due to the slowness

with which the valve closes the port. This is clearly exhibited in the diagrams of all engines fitted with such valves, by a rounded corner at the point of "cut-off." A perfect valve should open quickly and remain open until the point of cut-off, then close quickly. These conditions are not fulfilled by any of the valves in ordinary use, unless, perhaps, the Corliss and the Proel valve gears. The opening to steam should be sufficiently large to allow the steam to pass into the cylinder with a velocity and volume, greater than the displacement of the piston, if this drop in pressure is to be avoided.

Release.—In order to prevent excessive back pressure during exhaust, it is necessary to release the steam pressure on the piston before the end of the stroke. This has the effect of rounding the right-hand corner of the diagram, as shown by the line, *efg*;

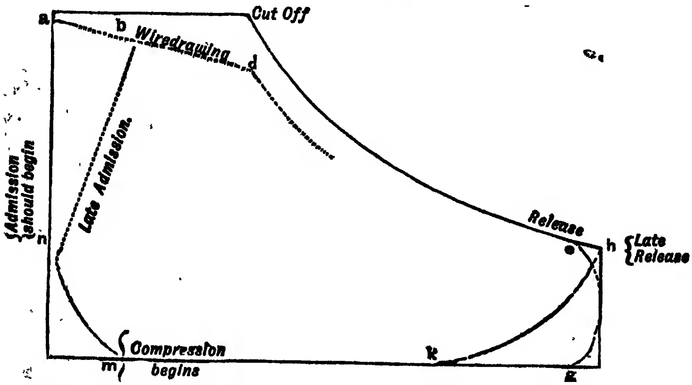


FIG. 6.—EFFECTS OF COMPRESSION, WANT OF LEAD, WIRE-DRAWING, AND RELEASE.

whereas, if steam be carried to the end of the stroke before exhausting, the diagram will take the form shown by the line, *h k*, whereby excessive and wasteful back pressure will be the result.

Compounding.—Having now studied the principal points in connection with the *actual* expansion of steam in a single cylinder, we are in a position to explain the several advantages of the compound engine over the simple expansive engine.

Advantages.—(1) From what has been already stated, the student will see, that the amount of liquefaction which takes place in a single cylinder engine varies with the difference between the initial and final temperatures. Therefore, the loss from liquefaction in any cylinder increases as we increase the

ratio of expansion. The principle of the compound engine, then, consists in dividing the expansion into two or more stages, and carrying out each stage of expansion in a separate cylinder, so as to reduce the variation of pressure and temperature in each cylinder. For some time there was great diversity of opinion as to the advantages of the compound system, but actual practice has now proved beyond doubt that, if properly proportioned, the compound engine is much more economical in consumption of fuel for a given power than the simple expansive engine. Not only is the amount of liquefaction reduced in the compound system, but any re-evaporation of condensed steam which may take place in the high-pressure or intermediate cylinders during exhaust is not a direct source of loss, for, although increasing the back pressure in these cylinders, it is not at once discharged into the condenser, but passes on to the next cylinder and does useful work there.

(2) The loss from clearance is also less in compound engines, for, as we have shown, the loss from this cause increases with the ratio of expansion in each cylinder.

(3) In the simple engine, with a high ratio of expansion, there is of necessity a wide variation of pressure on the piston. This causes an irregularity of rotational effort on the crank-pin, which is objectionable, and the initial stress (which all the working parts must be strong enough to withstand) is far in excess of the mean stress by the compound arrangement.

(4) With compounding, we may use very high initial boiler pressures and permit the steam to do work successively in two, three, or more cylinders placed in series. As previously explained, we obtain high-pressure steam at a comparatively small extra cost. For, as clearly shown by Table II., Lecture VII., the total heat units in 1 lb. of steam at 20 lbs. pressure is nearly 1,152 B.T.U.; at 100 lbs. it is 1,182 or only 32 more, and at 200 lbs. it is 1,198 or only 16 B.T.U. more than at 100 lbs. Now, by using steam of 200 lbs. initial pressure and letting it expand and do work in, say, three successive cylinders, we get far more out of each lb. of steam than we could, by using one big single cylinder with 20 lbs. initial pressure, and more proportionately than it costs to raise the steam from 20 to 200 lbs. pressure.

LECTURE XV.—QUESTIONS.

N.B.—The questions from 1 *to* 19 *are directly upon the text of this lecture and roughly in the order thereof. The following questions are in order of the dates when they were set, and the large number of A.M.I.C.E. questions, &c., bearing upon this lecture will be found in the Appendix.*

1. Account for the usual losses of pressure and temperature in steam between a boiler and its engine.

2. Why does initial condensation of steam take place in the cylinder of a steam engine?

3. Enumerate the most successful devices for preventing or reducing initial condensation of steam in an engine cylinder.

4. Trace what happens in the working of a steam engine when the cylinder is not provided with a steam jacket.

5. What is the object of a steam jacket? In what way does the absence of the jacket affect the indicator diagram? State how the economy of the steam jacket is affected by (1) cut-off, (2) size of cylinder, (3) revolutions per minute, and (4) superheat.

6. Give a concise historical statement of the introduction and appreciation of superheated steam.

7. Describe, by sketches and index to parts, a superheater for a land and for a marine boiler.

8. Draw and explain the seven different expansion curves described in this lecture.

9. State the law according to which superheated steam expands in volume when its temperature is raised under a constant pressure. When steam is superheated for the supply of an engine in the usual manner, does its pressure rise above that in the boiler? Explain fully.

10. Distinguish between *superheated* steam and *saturated* steam. According to what law is the pressure of superheated steam affected when it is compressed into a smaller space? What happens in the case of saturated steam?

11. Explain the difference between isothermal, saturation, and adiabatic expansion of steam, and draw carefully the curves for each in one diagram.

12. State clearly the real benefits derived from the use of superheated steam.

13. What is a separator and how does it act?

14. What is meant by the term "clearance?" Assuming that the clearance has been reduced to an equivalent length of the stroke of piston, which is 4 feet, and taking the case where steam is cut off at half-stroke, the clearance being 3 inches, you are required to compare the pressure of the steam, when 3 feet of the stroke are made, with the pressure under the same circumstances if there were no clearance. *Ans.* 27 : 26.

15. Define the terms "clearance" and "ratio of expansion" as applied to a steam engine. Draw a theoretical indicator diagram for a condensing engine, where the steam is cut off at $\frac{1}{4}$ stroke. Mark, on this diagram, in dotted lines and writing, the effects of (*a*) clearance, (*b*) late admission, (*c*) wire-drawing, (*d*) late release, and (*e*) too early compression. Steam at 30 lbs. initial pressure by gauge expands to 12 lbs. absolute at point of release. Find ratio of expansion, given clearance = 5 per cent. of stroke, and release taking place at 7 per cent. of stroke before the end. *Ans.* 4.

16. Explain by an indicator diagram the meaning of the terms "compression or cushioning," "lead," "wire-drawing," and "release." Show

how and why the work spent upon compressing the tail end of the exhaust steam up to the initial admission pressure is not given out completely as work done during expansion.

17. What are the principal causes for the presence of water in the cylinders of steam engines? What methods have been employed to diminish the loss due to initial condensation and subsequent re-evaporation in the cylinders? Hence state the advantages which result from the use of compound cylinder engines.

18. State and explain the several advantages due to compounding.

19. Plot neatly and accurately, in different colours and on squared paper, to a vertical scale of $\frac{1}{16}$ inch = 1 lb. pressure, and to a horizontal scale of $\frac{1}{10}$ inch = 1 cubic foot of volume, the isothermal, dry saturated, adiabatic, and superheated steam curves direct from their respective equations, with an expansion of ten times from the point of cut-off and with the initial pressure of 100 lbs. absolute per square inch. Also, plot down what you consider to be the probable curve for steam of 100 lbs. initial pressure with 20 per cent. initial condensation. Then find by measurement, and, by the rules adopted in Lecture XII., the mean pressures obtained from each of these curves, and thus check the table of percentages which these mean pressures are of the initial pressure.

20. Explain why condensation takes place in a steam engine cylinder. Explain carefully, giving figures if possible, the usefulness (and the limit to it) of expansion in one cylinder. Why and under what conditions is it advantageous to expand in two or more cylinders instead of in a single cylinder? (S. & A., 1897, Adv.)

21. Fluid expands from a point on the diagram where p is represented by 1·5 inches, and v by 1 inch, to a place where v is 3·5 inches. According to each of the laws of expansion, $p\,v$ constant, $p\,v^{1\cdot0646}$ constant, and $p\,v^{1\cdot13}$ constant, find the value of p at the end of the expansion in each case. (B. of E., 1900, Adv.)

22. Assuming no clearance; cut-off at one-third of the stroke; expansion according to the law, "$p\,v$ constant;" what is the mean forward pressure as a fraction of the initial pressure? If the cross-section of the cylinder is 144 square inches, length of stroke 2 feet, what volume of steam is used per stroke? If the back pressure is 17 lbs. per square inch and there are 200 strokes per minute, find in the following two cases the indicated horse-power and the weight of steam used per hour. Neglect clearance, condensation, and leakage.

Initial pressure in lbs. per square inch, . . .	180	100
Volume in cubic feet of initial pressure steam per lb.,	2·51	4·356

Use squared paper to show the weight of steam per hour used by the engine at any power. (B. of E., 1900, H., Part i.)

23. In the previous question, with initial pressure 180, find the mean forward pressure during a stroke. Neglecting the shortness of the connecting-rod, find the pressure when the crank makes angles of 0°, 15°, 30°, &c., with its dead point, and find the average of these. (B. of E., 1900, H., Part i.)

24. Without giving the mathematical investigation, state what is the result of our study of the cause of the initial condensation in a cylinder;

has it been confirmed by experiment? What is known about steam missing through leakage past valves? (B. of E., 1900, H., Part i.)

25. We endeavour to prevent condensation in the cylinder of a steam engine (*a*) by a separator, (*b*) by superheating, (*c*) by drainage from the cylinder, (*d*) by steam jacketing, (*e*) by high speed. Explain how each of these methods tends to affect our object. (B. of E., 1901, Adv. and H., Part i.)

26. In an air compressor the air is drawn in at a pressure of 14·7 lbs. per square inch absolute, and compressed to 77·0 lbs. per square inch absolute. The volume drawn in per stroke is 1·52 cubic feet, and when 133 strokes are made per minute the compressor requires 24·16 H.P. to work it. What H.P. would be needed if the air were compressed isothermally? Hence find the efficiency of the compressor ($\log_e 5{\cdot}24 = 1{\cdot}656$). (C. & G., 1901, H., Sect. B.)

27. What do you understand by cylinder condensation in a steam-engine cylinder? Explain the various devices adopted to reduce it. (C. & G., 1902, O., Sec. C.)

28. Explain, as fully as you are able, what is meant by cylinder condensation, and why it causes a loss of efficiency. Describe three ways by which it may be reduced, and discuss the advantages and disadvantages of each method (1) from a theoretical and (2) from a practical point of view. (C. & G., 1902, H., Sec. B.)

29. Show that no more work is obtained from a given quantity of dry steam by passing it dry through two cylinders, as in a compound engine, than by admitting it into a low-pressure cylinder only with the same degree of expansion, on the supposition that no heat is conducted away and radiated by the sides of the cylinders.

LECTURE XVI.

CONTENTS.—Watt's Indicator—Special Features of the New Crosby Indicator—Description of the Crosby Indicator—Errors in Indicators—Recording Mechanism—Taking of Indicator Diagrams—Examples of Defective Diagrams and the Causes of their Defects—Combined Compound Diagrams—Fairbairn's Saturation Curve—Graphic Representation on the Indicator Diagram of the Water present during Expansion—Gain in Pounds of Steam per I.H.P. due to Superheating—Gain in B.T.U. per I.H.P. due to Superheating, with Formula—The Effects of Raising the Superheat on the Indicator Card and on the Economy of Steam—Appendix to Lecture XVI. on Amsler's Planimeter—Questions.

Watt's Indicator.—Watt was the first who recognised fully the importance of gaining some knowledge of the action of steam in the steam cylinder of an engine, and the first form of indicator was the result of his efforts in that direction. The figure shows an improved form of Watt's indicator, by which a complete diagram could be traced out.

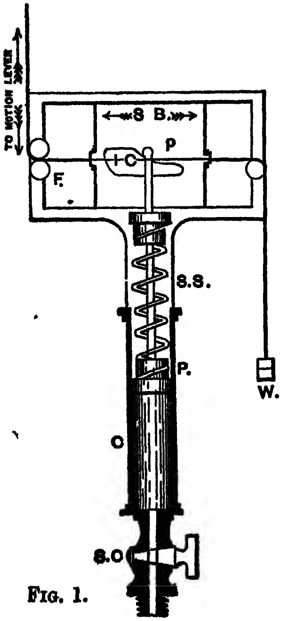


FIG. 1.

It consists essentially of a steam cylinder, C, about 1″ diameter and 6″ long, having a solid piston, P, accurately fitted into it. The cylinder is open at the top, and is fitted with a steam cock, S C, at the bottom, which is screwed into the cylinder cover of the engine, or into the engine cylinder itself close to the end. A small rod is fitted into the piston at one end, and carries a pencil, *p*, at the other, which can operate on a sheet of paper fixed on the sliding board, S B, in front of it. The sliding board can move horizontally in the frame, and receives its motion by means of a cord which is fastened to some reciprocating part

INDEX.

S C	for	Steam cock.
C	,,	Cylinder.
P	,,	Piston.
S S	,,	Spiral spring.
p	,,	Pencil.
F	,,	Frame (wood).
S B	,,	Sliding board (covered with paper).
I C	,,	Indicator card.
W	,,	Weight attached to cord for return motion of S B.

of the engine, the period of whose motion is identical with that of the piston of the engine. The return of the board is effected by means of the weight, W, and the cord, while the vertical motion of the piston is controlled by a spiral spring, S S.

When the instrument was first brought into use by Watt, the pencil moved in front of a graduated scale, but no lateral motion was given to the paper, hence, all the information obtainable was the pressure of the steam in the cylinder, or the perfection of the vacuum. The addition of the sliding board, however, enables a complete diagram to be set out, and the steam pressure and vacuum ascertained *at any point of the stroke.* The importance of this improvement will be at once apparent.

Different Kinds of Indicators.—There are a great number and variety of these useful instruments, but hitherto we have selected those which were generally recognised as being the most approved of their kind for special speeds and pressures. In previous editions of this book we have dealt with Richard's, Thompson's, and Wayne indicators. In present editions of the author's *Elementary Manual on Steam and the Steam Engine*, the Richard's and the M'Innes-Dobbie indicators are fully illustrated and described; consequently, we have now selected for explanation the latest American pattern of the high-speed Crosby instrument. (See Appendix E for the *Cipollina Indicator.*)

Special Features of the New Crosby Indicator.—1st. The spiral spring, S S_1, has been removed from the inside of the cylindrical case (near the piston, P) to the outside, and fixed above the moving parts, where it will remain cool under all conditions. Whatever error there was from the spring becoming heated in the ordinary indicator is not present in this instrument. The indicator is therefore suitable for taking indicator diagrams from an engine which may be supplied with superheated steam.

2nd. Another and more important difference lies in the size and shape of the piston. This piston is made 1 square inch in area, and is turned at its rubbing surface in the form of the central zone of a sphere. This greatly reduces the depth of the piston, which is still kept steam-tight by the thin film of moisture which collects around its lip. As the contact of the piston with the interior of the cylinder is a mere line, and the piston is attached directly by a rod to the upper part of the spiral spring, S S_1, it moves freely and without restraint inside the cylinder, notwithstanding the probability of some eccentricity in the action of the spiral spring.

3rd. The pencil mechanism is connected by a rod to and directly over the piston, by a ball and socket joint, B J. Consequently, as the piston takes up the torsional stresses of the

spring, $S S_1$, when it acts upon the pencil mechanism of the indicator, and its rod, P R, moves in a vertical line through a sleeve attached to the base of the pencil mechanism, the pencil point, P P, is compelled to move in a vertical line.

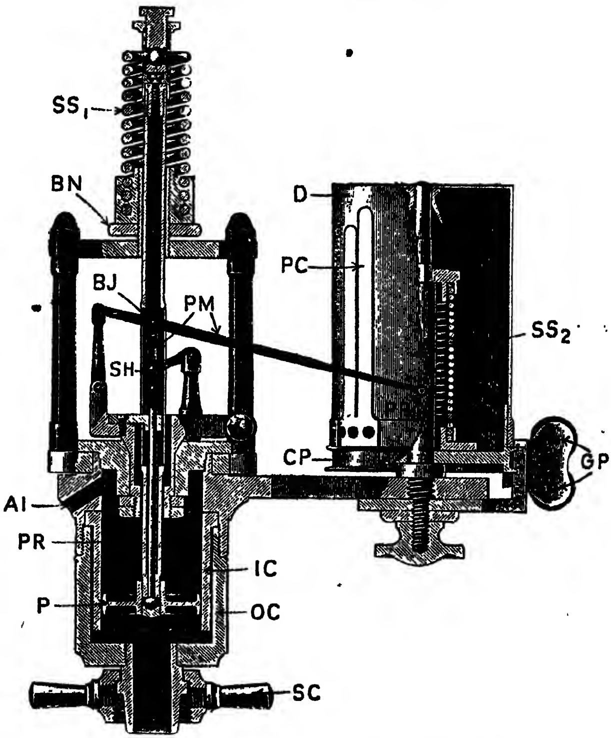


FIG. 2.—THE NEW SMALL CROSBY INDICATOR.

INDEX TO PARTS.

S C	for	Steam cock connection.	B N	for	Binding nut.
O C	,,	Outer cylinder case.	$S S_{1, 2}$	,,	Spiral springs.
I C	,,	Inner cylinder.	P M	,,	Parallel motion.
P	,,	Piston.	P P	,,	Pencil point.
P R	,,	Piston-rod.	P C	,,	Paper clips.
A I	,,	Air inlet.	D	,,	Drum.
S H	,,	Swivel head.	C P	,,	Cord pulley.
B J	,,	Ball joint.	G P	,,	Cord guide pulleys.

4th. Another feature is the adjustment of the pencil point, P P, to any desired position on the drum, D, by loosening the binding nut, B N, and screwing the spiral spring, $S S_1$, upwards or downwards. The spiral spring, $S S_1$, carries with it the whole pencil mechanism, P M. Having adjusted the pencil point, P P, to the desired position on the paper by means of the spiral spring, you again fix it by tightening firmly the binding nut, B N.

Description of the Crosby Indicator.—This indicator has a little barrel or outside cylinder, O C, which communicates with the engine cylinder and stop-cock through the steam-cock connection, S C. It also has an inner cylinder, I C, in which the piston, P, moves steam-tight, due to the condensed steam collecting around the lip of the piston and cylinder rubbing surface. Between the outside and the inside cylinders is an annular space, which acts as a steam jacket to the inner cylinder. It will also be noticed, that I C is perfectly free at its lower end, thus allowing it to expand and contract. The hollow steel piston-rod, P R, is screwed into the piston, P. The pressure of the steam in the engine cylinder raises the piston, P, extends the spiral spring, $S S_1$, and causes the pencil point, P P, to rise in a straight line through a distance on a magnified scale, proportional to the extension of the spring, and therefore to the pressure of the steam. During this upward movement of the piston, the swivel head, S H, and parallel motion levers, P M, are also raised. Hence, a line is thus traced by P P upon the paper or card, which is wound round the drum, D, and fixed by the paper clips, P C. This drum, D, rotates for about three fourths of a revolution and back again to its original position as the cord, cat gut, steel wire, or steel-wire strip (which is wound round the cord pulley, C P, near the bottom) is pulled and let go. The cord or wire from C P passes over the guide pulleys, G P, to any convenient form of reducing arrangement attached to the cross-head of the engine. Inside the drum, D, there is a spiral spring, $S S_2$, fixed at its lower end, and which becomes wound up when the cord on C P is pulled This spring, $S S_2$, serves to turn the drum in the reverse direction during its back or return stroke. An air inlet, A I, is provided in the cap of the indicator cylinder to admit air freely to the top of the piston. When the pressure of the steam in the engine cylinder is less than that of the atmosphere, the difference between these pressures acts on the top of the piston, P, causing it to descend and compress the spiral spring, $S S_1$. The tap placed just below the indicator at S C allows steam communication with the engine cylinder to be cut off at pleasure. It also permits the indicator cylinder to

be placed in direct communication with the atmosphere, thereby allowing the atmospheric line to be drawn on the diagram.

Spiral Springs.—The indicator springs are each made of one piece of steel wire, as shown by the separate view. They are right- and left-handed, and therefore have no tendency to press the piston laterally against the cylinder when either extended or compressed. Springs adapted to various ranges of steam pressure are supplied with each indicator, and are marked with a number which states the pressure in lbs. per square inch,

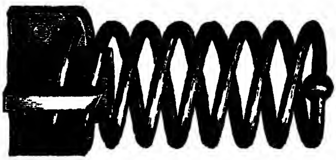


FIG. 3.—SPIRAL SPRING FOR CROSBY INDICATOR.

that will raise the pencil point, P P, through a distance of *one* inch on the paper attached to drum, D. In this form of indicator the spiral spring, S S_1, is kept away from the indicator piston and cylinder, and is well exposed to the atmosphere. Consequently, the spring will generally be about the same temperature as the atmosphere. At the same time it is easily accessible for changing the spring at any time.

Testing the Spiral Springs.—The accuracy of the number marked upon each spring, should be carefully verified by testing the indicator under steam, against a standard pressure gauge or by a mercury column. The spiral spring is found by experiment to be much stiffer when cold than at the higher temperature when in use. Therefore, water-pressure tests are not suitable, unless a proper allowance be made for the change of elasticity of the spring, through its change of temperature.

Errors in Indicators *—These errors may be due to one or more of the following defects:—

(1) In indicators where the spring is placed inside the cylinder, the stiffness of the spring alters with the temperature of the steam. Also, the average temperature of the spring is not known, and is different in every instance.

* The student who is desirous of investigating the different errors to which some indicators are liable, should study Prof. Osborne Reynolds' paper on "The Theory of the Indicator and the Errors in Indicator Diagrams;" also, "Experiments on the Steam Engine Indicator," by H. W. Brightmore, *Proc. Inst. C.E.*, vol. lxxxiii., Part i.

(2) Through defects in the parallel motion arrangement, P M, or in the spiral spring, S S_1, itself, the vertical motion of the pencil is not exactly proportional to the pressure in all positions.

(3) Bad mechanical fitting of the parts, either through bad workmanship or wear and tear of the instrument.

(4) The inertia of the drum or barrel, D, combined with weakness of spring, S S_2; or, the strength of spring, S S_2, combined with the yielding of the cord attached to the cord pulley and reducing arrangement on engine crosshead, sometimes cause too great or too little travel of the drum. In both cases, the motion of the paper on the drum, D, is not an exact reduction of the movement of the engine crosshead.

(5) Friction, whether at the several joints of the parts moved by the piston or between the pencil point, P P, and the paper.

(6) Even, when the cord which is to move the indicator drum, D, is connected to the engine crosshead or piston-rod in such a way, as to copy its motion correctly, the motion of the drum itself may become incorrect, because the length of the cord is not strictly constant.

(7) The inertia of the reciprocating parts should be a minimum.

Reducing Mechanism.—In the first place, it is necessary to have a reducing mechanism, which will give a sufficiently reduced and accurate copy of the engine piston's stroke to the motion of the drum, D. Many arrangements are used for this purpose, such as in some forms of pantograph, whereby a geometrical solution of the problem has been aimed at. It is, however, not unusual to find in actual trials, greater errors than would occur with simpler forms of gear, due to the multiplicity of joints in the mechanism. All reducing gears should be simple in construction, not liable to get out of order or deranged in any way, and should be so arranged, that the string may be led as directly as possible from the crosshead to the indicator.

Mr. Frederick Sargant has invented and patented an electrical device applicable to an indicator, whereby any number of indicators can be operated, and diagrams taken at the same instant of time by closing an electric circuit.

The following figures show several different forms of reducing mechanism, of which Nos. 1, 3, 5, and 9 are simple and good; but the student should note and compare them all, by actual trial:—

It is impossible, within the limits of a general "Text-Book on Steam and Steam Engines" of this description, to enter very fully into the many devices, with their respective errors, for reproducing to a convenient scale the movement of the pistons

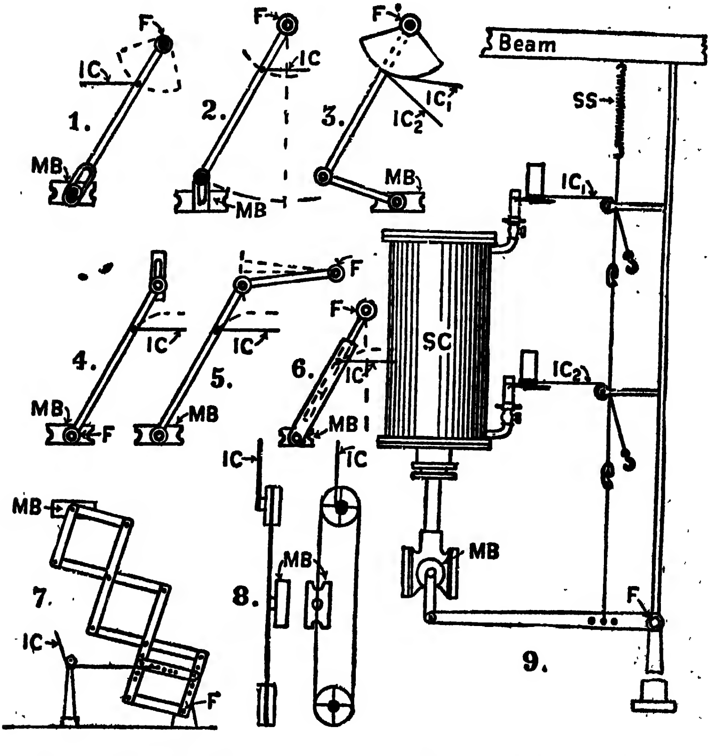
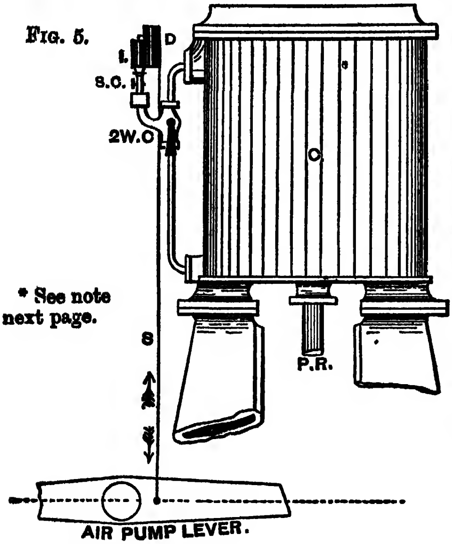


FIG. 4.—DIFFERENT FORMS OF REDUCING MECHANISM FOR STEAM ENGINE INDICATORS.

of steam engines. Those who desire to study this subject thoroughly, should refer to *Indicator Diagrams with Engine and Boiler Testing*, by Charles Day, Wh.Sc., of the National Boiler Insurance Company, Manchester.

In most engines a small pipe is fixed outside the cylinders and communicating with both ends.* The indicator is attached to this pipe. The pipe is fitted with a two-way cock, so that a diagram may be taken from either end of the cylinder at pleasure. This Fig. 5 shows the method of attaching the indicator to an inverted cylinder marine engine.

FIG. 5.



* See note next page.

The string or steel wire, S, is attached to the air-pump lever, and its travel must be rather less than the circumference of the drum, D. Before admitting steam into the indicator the "atmospheric line" should be drawn. This is done by turning the steam cock, S C, so that the indicator piston is put into direct communication with the atmosphere through a small hole, and then bringing the arm which carries the pencil up to the rotating *drum*, when a horizontal line is drawn. This line is marked A L, for "Atmospheric Line," on the diagrams throughout this book.

Indicator Diagrams.—Having studied in the previous lecture various effects produced on the theoretical indicator diagram by clearance, lead, compression, release, and such other arrangements as are required in practice for the proper expansion of steam in the cylinder of an engine, we are in a position to examine and comment upon a few indicator diagrams taken from actual practice.

The annexed diagram is taken from a horizontal non-condensing engine (sometimes wrongly termed a high-pressure engine), and, as will be seen, the diagram is exceptionally good. The steam pressure rises almost instantaneously, as shown by the vertical admission line, and is well sustained up to the point of cut-off, the line P C O being perfectly horizontal. At the point of cut-off, C O, a very slight wire-drawing may be seen by the rounded corner, but it is very inappreciable and testifies to the

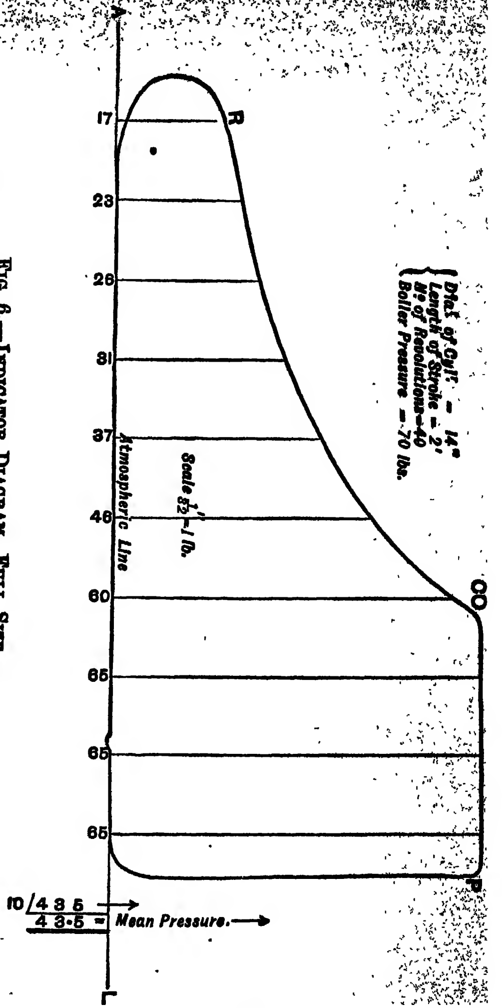


FIG. 6.—INDICATOR DIAGRAM, FULL SIZE.

This diagram was taken by the author from the Clyde Trust pumping engines working their Armstrong Hydraulic System at the Queen's Dock, Glasgow.

NOTE.—The plan of attaching the Indicator to both ends of a cylinder, as shown in the figure on last page, although convenient from a mechanical point of view, is not advisable in the case of long cylinders, or where the pipes are exposed to the cooling action of draughts. To obtain accurate diagrams, the Indicator should be attached directly to each end of the cylinder by a short large pipe, so as not to throttle or condense the steam.

efficiency of the valve gear. The release of the exhaust steam takes place at the point R, but might, with advantage, have been effected a little sooner. The exhausting of the steam is very effectually carried out, as the back pressure falls quite down to

the atmospheric line, A L. The amount of compression shown is too little, and a larger compression would no doubt make the engine work more smoothly at the dead points, for a slight knocking was observable. In this engine, however, the piston speed is very slow, viz., 160 feet per minute, so that a large amount of compression is not necessary.

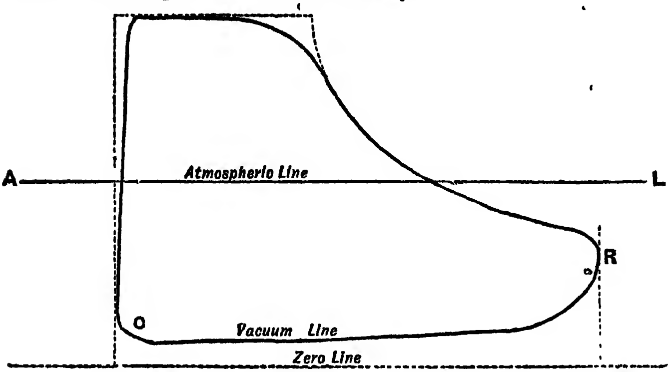


FIG. 7.—DEFECTIVE DIAGRAM FROM A SIMPLE CONDENSING ENGINE.

Defective Indicator Diagrams.*—Fig. 7 is a reduced copy of an actual diagram from a condensing engine. It presents one or two defects which we shall notice briefly. First, the amount of compression is too small and the valve has not had sufficient, if any, lead. The absence of proper cushioning is shown by the very small rounded corner at the point O, and the sloping away of the admission line from the vertical shows that the valve has not been sufficiently open when the piston reached the end of its stroke. Had the valve been set to give more lead, the admission line would have coincided with the vertical dotted line, and it is evident that the non-coincidence of these lines cannot be due to wire-drawing in the steam passages. For, when once the full pressure comes on the piston, it is fully sustained (as shown by the horizontal steam line) until the valve approaches the point of cut-off, when the usual wire-drawing takes place, due to the slow motion of the slide valve, and is clearly shown by the rounded corner on the diagram. Since this diagram is taken from a condensing engine the steam exhausts into a condenser, and the back pressure or vacuum line falls far below the atmospheric line, A L, but release has been given rather late as shown at R, and the exhaust shows contracted opening or wire-drawing, for it slopes down towards the left hand of the figure with gradually diminishing back pressure.

* Students who desire to study the many defects to be found in indicator diagrams and their causes should refer to *Indicator Diagrams with Engine and Boiler Testing*, by Charles Day, Wh.Sc., Published by The Technical Publishing Co., Manchester. Also, *The Steam Engine Indicator and Indicator Diagrams*, by W. Worby Beaumont, M.Inst.C.E., The Electrician Series; and Reed's *Engineers' Handbook*.

The next four examples of defective cards are taken from a question set at the February, 1900, A.M.I.C.E. Examinations of The Institution of Civil Engineers, where the candidates are asked to point out what is amiss with these four indicator diagrams, and to state the causes of the defects.

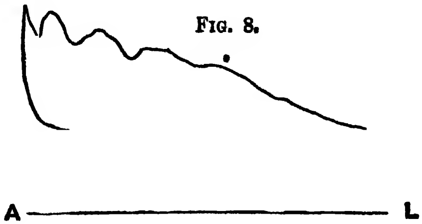


Fig. 8.

Fig. 8 shows up and down oscillations during the time of steam admission which die off gradually during expansion. These oscillations occur when the ratio of the speed of the engine to the stiffness of the indicator's reciprocating steam spring exceeds a certain value. Sometimes these oscillations may be caused or intensified by dirty indicator pistons or friction at a certain part of their stroke. They are, therefore, solely due to want of stiffness in the indicator spring for the speed and the momentum of the moving parts of the indicator. They might be damped by using stiffer springs, but then the diagram might be too small in height; or they might not occur if the moving parts of the indicator were made as light as possible. It is with this object in view that the moving parts in the Crosby and other indicators for indicating fast speed engines are made as light as possible. It will also be observed that the exhaust line is high above the atmospheric line, A L, thus showing that the diagram was taken from the first or high-pressure cylinder of a triple-expansion engine Further, the expansion is carried out to an extreme extent since the pressure in the cylinder falls to that of the back pressure before the end of the stroke.

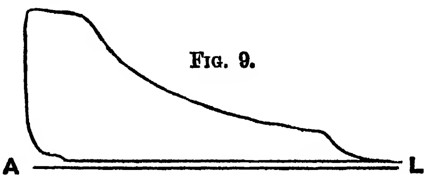


Fig. 9.

Fig. 9.—This diagram is evidently taken, either from a non-condensing engine, or from the H.P. cylinder of a compound engine, since the exhaust line is slightly above the atmospheric line. In either case, the release apparently takes place too soon, as shown by the hollow droop in the right-hand toe of the diagram. This defect may, however, be caused by want of clearance between the parallel motion lever and the curved arm of a Richards indicator. Also, the card would be improved if compression took place sooner. The sudden rise (so) or hiatus at the commence-

ment of the compression curve, may be due, either to a slight leak in the piston or the slide valve of the engine at this point of the stroke.

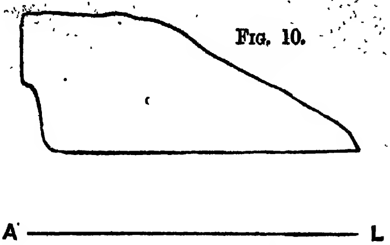


FIG. 10.

Fig. 10.—The *first* peculiarity to be observed in this diagram, is the irregular wavy admission line. This is probably due to the indicator pencil pressing too hard or unevenly upon the drum paper. The second fault shows a bad expansion curve which may be due, either to evaporation of initially condensed steam or to a leaky admission valve. The third and chief defect is seen at the compression corner of the diagram. This is evidently due to a leaky piston. This diagram is obviously obtained from the H.P. cylinder of a compound or triple-expansion engine, as the exhaust line is so high above the atmospheric line, A L.

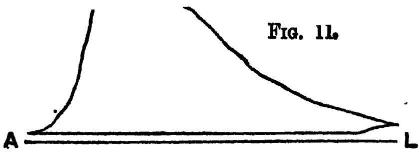


FIG. 11.

Fig. 11.—Looking at the left-hand end of this diagram we see, that there was neither cushioning nor lead. Both of these defects could be remedied by putting the eccentric sheave further forward, so as to give the valve more lead and cut off the exhaust sooner. We also see, that the exhaust release takes place too late, and this would also be remedied by the increased forward angle given to the eccentric. In fact, all the four points of compression, admission, cut-off, and release take place too late. If the engine is a non-condensing one, the back pressure is higher than it need be with a good free exhaust.

Turning now to Fig. 6 in the Appendix to this lecture, we see a very peculiar loop diagram. This kind of card is often obtained from the high-pressure cylinders of Corliss or drop-valve engines, when running on a very light or no external load. The exhaust port is closed almost from the commencement of the return stroke. Cushioning thus takes place to such an extent, that the back pressure rises above that of admission before the commencement of the next stroke, which causes the negative loop on the left-hand upper corner. The method of measuring, by Amsler's integrator, the net area of work done in this case should be studied.

Compound Engine Diagrams.—Diagrams from the cylinders of a compound engine should all be taken at the same time, so that the conditions of boiler pressure, cut-off, &c., under which each diagram is taken, may be the same. Since the pressure of steam in each cylinder is different, springs of different strengths are used in each indicator, and hence the diagrams of different cylinders are all to different scales. From these separate diagrams, therefore, we cannot get much information, except as regards the working of the valves and the amount of work developed by each cylinder. In order to tell accurately the extent of the loss of pressure between each cylinder, and the loss from liquefaction in the cylinders during admission, as well as any abnormal increase of pressure at any point due to re-evaporation or steam jacketing, we require to draw all the diagrams down to the same scale, when the distribution of steam may be clearly seen.

The saturation expansion curve should be plotted out on the same sheet of diagrams, so as to show clearly the variation of the actual expansion from this curve. In combining compound engine diagrams, it is best to take the volume of steam undergoing expansion, as equal to that of one pound of steam at the given pressure, so that all diagrams shall be drawn on the same basis and may be compared with each other If we do this, we can readily construct the saturation expansion curve from tables without any calculation or geometrical construction. A diagram of the expansion of dry saturated steam is plotted in the turbine lecture, and greatly facilitates the construction of the saturation curve. In this diagram the vertical ordinate represents absolute pressures in lbs. per square inch, while the horizontal abscissa represents the volume in cubic feet of one pound of steam. By its aid—if the volume of steam undergoing expansion is one pound—we can read off the pressure corresponding to any particular volume; and, if we set off this pressure at several different points throughout the stroke, we have only to join those points in order to complete the saturation curve.

To illustrate this important point, we append the diagrams of the compound engines of H M.S. *Boadicea*, and proceed to show how to reduce them to the same scale and draw the saturation curve. The engines have one high-pressure and two low-pressure cylinders, and the ratio of the joint capacity of the two low-pressure cylinders to the high-pressure cylinder is 3·11 : 1. The steam is cut off at ·46 of the stroke in the high-pressure cylinder, and the volume of the high-pressure cylinder is 116·25 cubic feet. The pressure of the steam is 80 lbs. absolute, and the clearance of each cylinder $\frac{1}{11}$ of the volume of the cylinder.

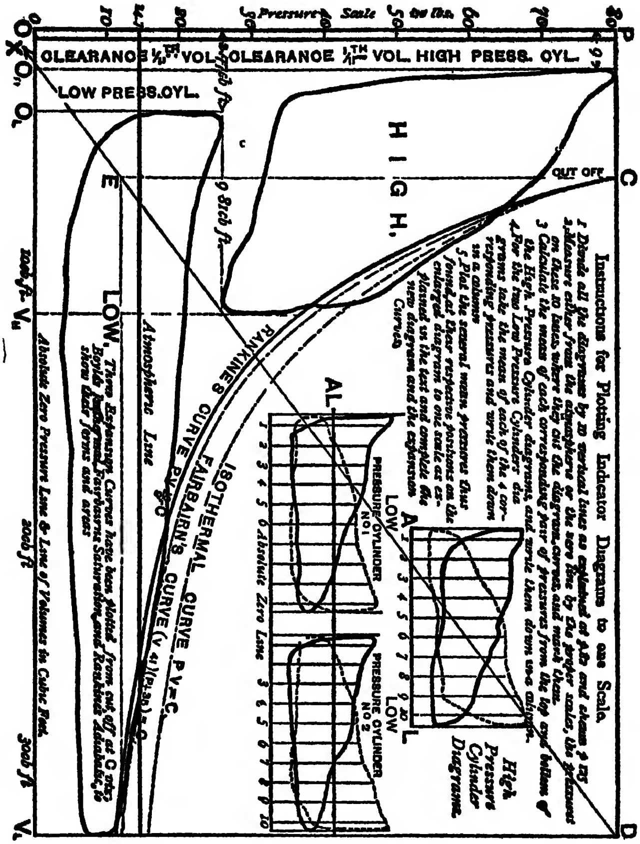


FIG. 12.—COMBINED INDICATOR DIAGRAMS—H.M.S. "BOADICEA."

Quantity of steam used in high-pressure cylr. } = volume of cylr to pt. of cut-off + clearance.

$$\text{"} \quad \text{"} \quad \text{"} \quad = \text{vol. of cylr.} \times \cdot 46 + \frac{\text{vol. of cylr.}}{11}$$

$$\text{"} \quad \text{"} \quad \text{"} \quad = 116 \cdot 26 \times \cdot 46 + \frac{116 \cdot 26}{11}$$

$$\text{"} \quad \text{"} \quad \text{"} \quad = 64 \cdot 04 \text{ cubic feet.}$$

The volume of one pound of steam at 80 lbs. pressure may be found from Rankine's diagram already referred to, or the table in Lecture XV., to be 5·4 cubic feet.

$\therefore$ Weight of steam used in high-pressure cylinder in each stroke $\Big\} \; \dfrac{64·04}{5·4} = 11·85$ lbs.

$\therefore$ Volume of high-pressure cylr. per lb. of steam (without clearance) $\Big\} \; \dfrac{116·26}{11·85} = 9·81$ cubic ft.

Clearance of high-pressure cylr. per lb. of steam used $\Big\} \; \dfrac{9·81}{11} = ·9$ cubic ft. nearly.

Volume of low-pressure cylr per lb. of steam (without clearance) $\Big\} \; 9·81 \times 3·11 = 30·5$ cubic ft.

Clearance of low-pressure cylr. per lb. of steam used $\Big\} \; \dfrac{30·5}{11} = 2·77$ cubic ft.

We are now in a position to construct the diagram. Lay off to scale the line, $O\,V_L$, equal to the volume of the low-pressure cylinder per lb. of steam + its clearance = 30·5 + 2·77 = 33·27 cubic feet, and draw the vertical line, O P, to represent to scale the initial pressure of 80 lbs. per square inch. Measure off $O\,O_H$ = ·9 cubic feet, and draw a vertical line through O_H; this represents the clearance of the high-pressure cylinder. Now, make $O_H\,V_H$ = the volume of the high-pressure cylinder per lb. of steam, and divide this space into 10 parts, to correspond exactly with the divisions on the actual indicator diagram. Lay off on these divisions the mean pressures shown by the indicator diagrams, and complete the diagram of the high-pressure cylinder. The diagram of the low-pressure cylinder is reduced in the same way. $O\,O_L$, represents the clearance, and $O_L\,V_L$, the volume of the cylinder per lb. of steam, and in measuring pressures the mean of the 4 low-pressure cylinder indicator diagrams is taken.

The construction of the saturation curve from Rankine's formula $P\,V^{\frac{17}{16}}$ = constant, or Table II, is extremely simple, since we are dealing with one pound of steam, and the pressure corresponding to any particular volume may be set down at once.

Having now completed our diagram, we have a clear insight into the actual working of the steam in the cylinders of the engine. Evidently a large amount of wire drawing takes place in the high-pressure cylinder, as is shown by the great fall of pressure before the point of cut-off. The rise of pressure above the saturation curve which takes place during expansion, may partly be accounted for by the action of the steam jacket in re-evaporating

moisture in the cylinder, or by the steam which condensed on admission re-evaporating due to getting heat from the cylinder liner and the piston, &c.* The loss of pressure between the two cylinders is from 5 to 8 lbs., and is rather more than we usually find in well-designed engines. Wire-drawing also takes place in the low-pressure cylinder, although not to such a great extent as in the high-pressure cylinder.

Fairbairn's Saturation Curve.—When sufficient data cannot be obtained to reduce the volume of the saturation curve to that of one pound of steam, the diagrams may be drawn to the same scale, and the saturation curve drawn from the point of cut-off, C, by a simple geometrical construction. This is founded on the formula of Fairbairn and Tate for the expansion of saturated steam given in the last lecture, $(v - 41)(p + \cdot 35) = \text{constant} = 389$.

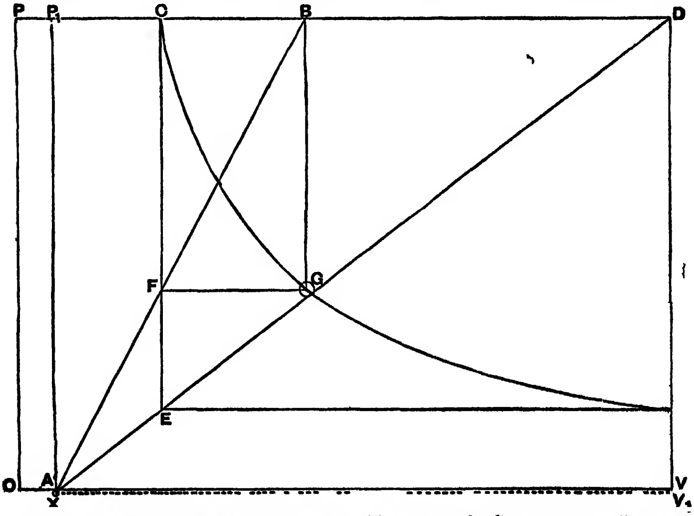


FIG. 13.—METHOD OF CONSTRUCTING FAIRBAIRN'S SATURATION CURVE.

The diagram shows how this may be done. Lay off O A on the line of volumes equal to ·41 cubic foot, and from A measure A X off below O V = ·35 of a pound pressure, and draw new axes of pressure, and volume X P_1 and X V_1, and on these axes describe an hyperbola. The hyperbola considered

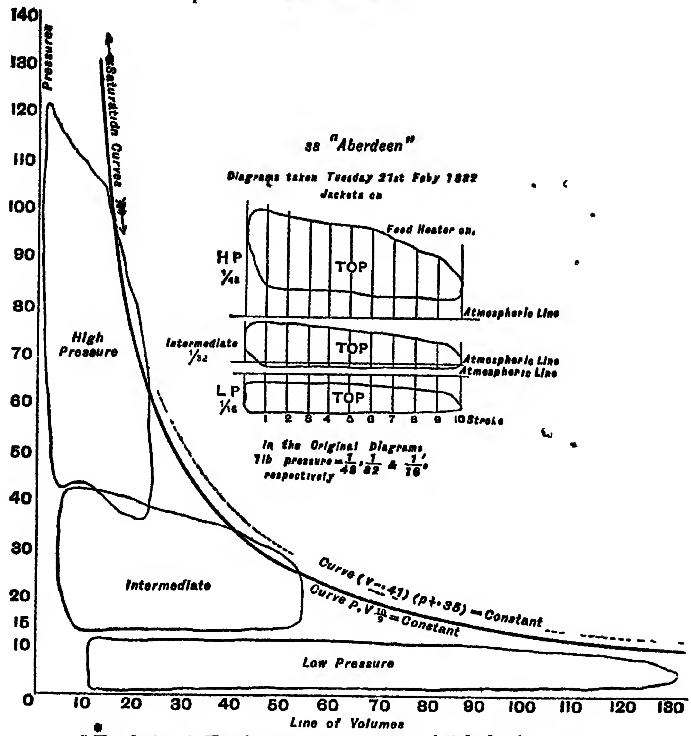
* Steam when passed through contracted and tortuous passages, so as to be reduced in pressure or "wire-drawn," becomes more or less superheated by the friction of its molecules, heat being the direct result of the work spent on friction. This may also account for some of the rise.

relatively to the old axes will be the curve of saturated steam, and, although it does not coincide *exactly* with Rankine's curve, it is sufficiently near for all practical purposes. To find points on the hyperbolic curve, the following construction is the simplest:—Take any point, B, between C and D, and join it with X. Where B X cuts the vertical line, C E (through the point of cut-off, C), draw a horizontal line, F G, cutting the perpendicular let fall from B to G, then G is a point on the curve. By finding a number of points in this way, the whole saturation curve may be drawn as in the next figure.

Combined Diagrams of a Triple-Expansion Marine Engine.—We pointed out in a former lecture, that the principal advantage of compound over simple expansive engines is, that the cylinders are not subjected to such great variation of temperature, and therefore, the loss from liquefaction in the cylinders is less. With steam pressure of from 60 to 100 lbs. in compound engines, the expansion is carried out in two cylinders only, but when that pressure is exceeded, the difference of the initial and final temperatures of the steam in each cylinder becomes so great, that three or more cylinders are required to expand the steam efficiently. The diagrams from the engines of the S.S. "*Aberdeen*," designed by Mr. A. C. Kirk, of Messrs. Robert Napier & Sons (which were among the first triple expansive engines constructed), are shown on the previous page drawn to the same scale. These diagrams show that there is very little loss of pressure between the cylinders, and fit in very well with the expansion curve $P V^{\frac{10}{9}}$. Very little re-evaporation takes place since the range of temperature in each cylinder is small.*

* The S.S. "*Aberdeen*" is an iron ship, built in 1881 for Messrs. George Thompson & Coy.'s London-Australian trade, by Messrs. Robert Napier & Sons, Glasgow, to the highest class at Lloyds—350 ft. by 44 ft. by 33 ft. The engines were supplied with steam at 125 lbs. pressure from two ordinary double ended boilers, with no superheater, constructed entirely of steel, with six of Fox's corrugated furnaces in each, the total heating surface being 7,128 square ft. The cylinders were three in number, being 30 in., 45 in., and 70 in. diameter respectively, by 4 ft. 6 in. stroke. The high-pressure cylinder was not steam jacketed, the second was steam jacketed, with steam of 50 lbs. pressure; and the low-pressure one with steam of 15 lbs. above the atmosphere. On the official trial 2,000 tons of dead weight were put on board, and a test made for the consumption of coal on a six hours' run at 1,800 I.H.P. The result was a consumption of 1·28 lbs. of Penrikyber Welsh coal per indicated horse-power.

See *The Proceedings of the Institution of Naval Architects*, 1882, for Mr. Kirk's paper "On the Triple Expansion Engines of the S.S. '*Aberdeen*,'" and for a paper "On the Economy of Compound Engines," by W. Parker Chief Engineer of Lloyd's Register, with discussions thereon.



* Fairbairn & Tate's curve. † Adiabatic curve.

FIG. 14.—COMBINED INDICATOR DIAGRAMS FROM THE TRIPLE-EXPANSION ENGINES OF THE S.S. "ABERDEEN."

Graphic Representation on the Indicator Diagram of the Water present during Expansion.—Professor Ewing in his book on *The Steam Engine* states, that the whole quantity of *steam and water present during expansion*, is the cushioned steam *plus* the "cylinder feed." The quantity of steam which passes through the cylinder per stroke is the weight of steam admitted during each stroke up to the point of cut-off, which he terms the "cylinder feed."

To estimate the amount of cushioned steam, he takes on the indicator diagram a point after compression has begun (*i.e.*, after the exhaust valve has become completely closed), and he notes the pressure and the volume there, remembering that the true volume is the sum of the incompleted portion of the stroke plus the clearance. From this pressure and volume

the quantity of the cushioned steam is readily calculated, *assuming that the steam is simply saturated and that no water is present when compression begins.* As a rule, this assumption is probably correct. Occasionally the cushioned steam may be wet (which would make its weight or amount greater), but in most cases the supposition that the steam is dry when compression begins, may be accepted as involving at least no serious error. The total quantity of steam in the cylinder during expansion is next found by adding the amount of this cushioned steam to the "cylinder feed." A dry saturation curve ($PV^{\frac{17}{16}}$ = constant) can then be drawn on the indicator diagram, to show the volume which this total quantity would fill *if* it were dry and saturated, at each pressure reached during the expansion.

For example, take the following reduced diagram which he took from a small engine of the marine type. Here the line, S S, is the dry saturation curve, which is drawn with the ordinate 0 to 60 lbs. as its origin, to the left of the diagram which the indicator traced, by a distance which represents the volume of the cylinder clearance.

If a horizontal line, A B S, be drawn to intersect the expansion curve at any point B, then A B represents the actual volume which the expanding mixture filled at the pressure O A; and A S is the volume which it would have filled had it been dry saturated steam, whilst B S represents the volume that is lost due to wetness. Hence, the proportion of water in the mixture is sensibly $\frac{BS}{AS}$, and the dryness fraction $x = \frac{AB}{AS}$. Thus, the proportion of water present at any stage of the expansion may be similarly determined.

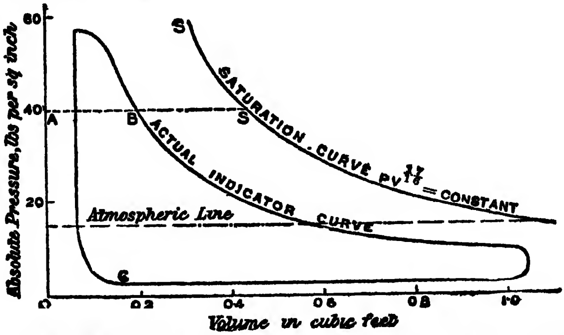


FIG. 15.—REPRESENTING THE QUANTITY OF STEAM AND WATER PRESENT DURING EXPANSION IN A SIMPLE CONDENSING ENGINE.

In the engine in question he found, that the amount of "cylinder feed" per single stroke was 0·0404 lb. The pressure at the compression point, C, was found to be 4 lbs. per square inch absolute and the volume was 0·12 cubic foot. Since the volume of 1 lb. of steam at that pressure of 4 lbs. is 90·4 cubic feet, it follows, that the amount of cushioned steam was 0·0013 lb. This gives a total of 0·0417 lb., for which the saturation curve, S S, was drawn. By measuring values of B S/A S at points along the curve, it was found that the proportion of water in the mixture was 52 per cent. at cut-off, then increased to about 55 per cent. during the early stages of expansion, became less, and finally sank to 37 per cent. just before release took place.

The student will note from this actual case, that in small unjacketed engines the amount of water present in steam at and immediately after the point of cut-off may be fully 50 per cent. of the weight of dry steam taken from the boiler. This fact is not generally recognised or understood by steam users, but it most undoubtedly accounts for the great benefits derived from so superheating steam, that it shall remain *perfectly dry* up to the point of cut-off, or even to the end of the stroke.

When dealing with compound engine diagrams, Prof. Ewing says, it is better to modify the construction of the previous figure by separating the cushioned steam from the cylinder feed and drawing the diagram for the latter. The reason for this modification is, that the amount of cylinder feed is the same for both or all the cylinders, whereas the amounts of cushioned steam may be different in each cylinder. This allows a combined diagram to be drawn for the several cylinders along with one saturation curve.*

It will be observed, that this new method has not been followed in the combined previous curves of H.M.S. "Boadicea" and S.S. "Aberdeen;" for *there*—as is usual in ordinary marine practice—the saturation curves have been drawn on the assumption that the steam was dry at the point of cut-off; and further, that the amount of substance which is taking part in the expansion is the same in the different cylinders. Consequently, a single saturation curve cannot properly apply to all the cylinders unless the above method be followed. In fact, the proper position of the saturation curve for each cylinder of these two engines should be further to the right hand by the amount of liquified steam in each cylinder.

Gain in Lbs. of Steam per I.H.P. due to Superheating.†—It will

*See *Proc. Inst C.E.*, vol. xcix., 1889-90, for Prof Osborne Reynolds' paper on "Tests of the Triple Expansion Engines" at Owens College, Manchester.

†I am indebted to Mr. E A Reynolds, M A., of Messrs Willans & Robinson's Scientific Staff, Rugby, for the original curves from which Figs. 16 and 17 have been reproduced, and to his paper on "The Economy of Superheated Steam" recently read and discussed before the Rugby Engineering Society for certain data.

Data	Simple Non condensing Engine.	Compound Condensing Engine.	Triple Condensing Engine.
Indicated horse-power,	17·15	346	315
Mean pressure in lbs. reduced to the low-pressure cylinder,	38·64	50	35·55
Revolutions per minute,	450	350	360
Steam pressure in lbs. by gauge,	65	154	162
Cut-off,	·3 to ·45	...	...
Vacuum in inches,		27	26·2

be both interesting and instructive at this stage to consider the actual gain in the weight of steam required by a certain class of engine due to superheating the steam to different degrees F. above that of dry saturated steam of the same pressure. It will be seen from an examination of the three curves in Fig. 16

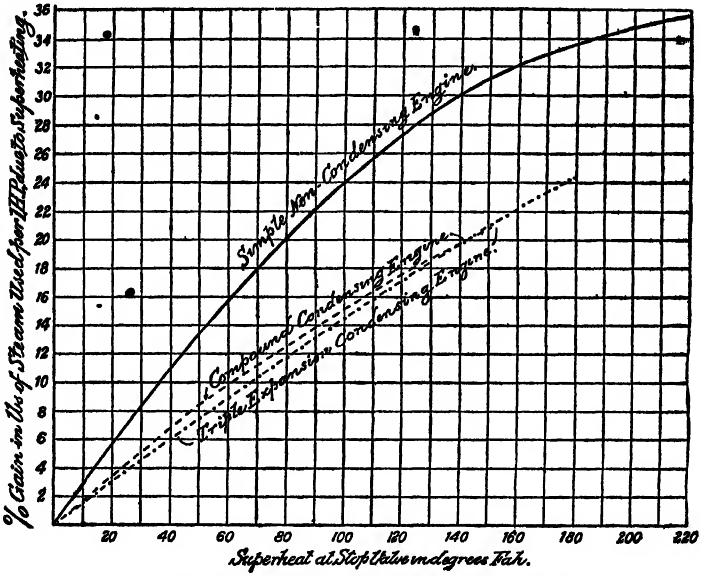


FIG. 16.—CURVES SHOWING THE PERCENTAGE GAIN IN FEED-WATER, OR STEAM USED PER I.H.P.-HOUR DUE TO SUPERHEATING THE STEAM, WITH MESSRS. WILLANS & ROBINSON'S SIMPLE, COMPOUND, AND TRIPLE-EXPANSION ENGINES.

that the percentage gain in the feed-water supplied to the boiler or in steam used, increases much more rapidly with the simple non-condensing engine (up to a certain degree of superheat) than with either the compound or the triple-expansion engine. The author is sorry that he has not got the data for the simple condensing engine. It is evident, however, that the curve for such an engine would lie on the diagram somewhere between that of the curves for the simple non-condensing and the compound-condensing engines; because, it might be taken as a general rule, that the greater the economy which an engine showed without superheating, the less would be the percentage gain by aid of superheating.

It will be observed from the inclination of the curves, that a quicker increase of gain was obtained at the lower degrees of superheat than at higher temperatures. This leads to the conclusion that there is not much gain *as a whole* by superheating steam to a higher degree before it enters a cylinder, than will just enable it to exhaust in a *dry condition* from that cylinder. Consequently, it would appear from this fact, and also from the other circumstances to be referred to later on, that instead of applying such a high degree of superheat as, say, 200° F., to high-pressure steam before it enters the first or high-pressure cylinder of multiple-expansion engines, it would be better to simply superheat at first, by 100° to 150° F., and then to reheat the exhaust steam from each cylinder by just the required amount; except, of course, the last or low-pressure exhaust, which is in connection with the condenser. From Mr. Reynolds' tests it appears, that very little difference in percentage gain was obtained with triple-expansion over that of the same class and power of compound engines with the same initial steam pressures and the same superheats. The gain in each case varied, of course, with the point of cut-off, or ratio of expansion. But, taken generally and roughly, it appears, that for a fixed cut-off in all the cylinders, the consumption lines at different degrees of superheat form a series of convergent straight lines, as shown by Fig. 16. Under these circumstances, it may be considered, that a simple non-condensing engine using superheated steam, could be made to work as economically as a condensing one at the same revolutions and power, when supplied with dry saturated steam. Also, a simple condensing engine should be equal to a compound one, and that it would be scarcely worth while to employ triple-expansion engines as far as economy, simplicity, and sweet working was concerned, when their extra complication, first cost, and upkeep was taken into consideration. If the superheat were high enough to let the steam be still dry at the exhaust of the intermediate cylinder of a triple-expansion engine, then the same economy in steam could be obtained by using a compound engine with a correspondingly early cut-off to give the same expansion. Of course, with the triple, there would be less exchange of heat between the metal of the cylinders and the steam than in the compound engine, due to the smaller range of temperature in each of the three cylinders. This would, however, entail perhaps an inconveniently high initial temperature in the first cylinder, and hence, as we said before, it would be better to reheat the steam in the intermediate receiver. This may be done by passing live, highly superheated steam, through

a coil fixed in the intermediate receiver on its way to the first cylinder steam-chest, which it would enter at a conveniently lower degree of superheat.

Gain in B.T.U. per I.H.P. Due to Superheating.—Results given in lbs. of water per I.H.P.-hour when using superheated steam are misleading, from the fact, that such a statement does not take into account the extra heat units imparted to the steam by superheating it. It has been suggested that a better comparison would be the number of lbs. of coal burned in the boiler furnace per I.H.P.-hour. But, it is well known, that coal varies much in calorific value, and boilers in efficiency. Consequently, this common but somewhat rough and ready method should be discarded when accurate and scientific comparisons have to be made. A more exact method would be to give the total heat units supplied to the water per I.H.P.-hour. In applying this method it is generally assumed, that the feed-water is at, say, 100° or 200° F. These are, however, mere arbitrary feed-water temperatures, which might be specially applicable to certain installations, but could not be recognised as fixed standards. The author, however, believes, that if the results were reckoned in B.T.U. supplied to the feed-water from 32° F. or from 212° F., a fair and uniformly applicable start could then be made from one or other of these two fixed temperatures. It would be most convenient to start from water at the higher fixed temperature of 212° F, because, as we saw in a previous lecture, the evaporative efficiency of boilers is reckoned by the lbs. of water which they generate into steam from and at 212° F.

Taking the case of the simple non-condensing Willans & Robinson's engine, it was found that when using steam of 65 lbs. pressure per square inch by gauge, or 80 lbs absolute in the steam chest, with a cut-off at ·3 of the stroke, a gain of 35 per cent. in the weight of steam resulted by superheating it 200° F., with a consumption of only 20 lbs. of steam per I.H P.-hour. (See the uppermost curve in Fig. 17.) Now, if 35 per cent. were the gain in this case, due to superheating, what would be the lbs. of steam per I.H.P.-hour, at the same pressure, cut off, and revolutions per minute, when supplied with ordinary dry saturated steam? Here, 100 per cent. – 35 per cent. (gain) leaves 65 per cent. used when superheated, what weight of steam would be required when it was saturated? Hence—

(Superheated) : (Saturated) : : (Weight superheated) : (Weight saturated).

$$65\% \quad : \quad 100\% :: \quad 20 \text{ lbs.} \quad : \quad x \text{ lbs.}$$

$$\therefore \; x = 30 \cdot 8 \text{ lbs.}$$

At 80 lbs. pressure absolute, reckoned from 32° F., the number of B.T.U. per lb. of this steam is 1,177 (see Steam Table II.). Subtracting from this total the sensible heat units per lb. of feed-water between 32° F. and 212° F. we get (1,177 - 180) = 997 B.T.U. This quantity multiplied by 30·8 (the lbs. of steam required per I.H.P.-hour), gives 30,707·6 as the total B.T.U. from and at water of 212° F. But the steam was superheated by 200° F., and assuming the specific heat of such steam to be 0·48; then (0·48 × 200) = 96 B.T.U. per lb., which, if added to the above 997, gives 1,093 B.T.U. per lb. of superheated steam.

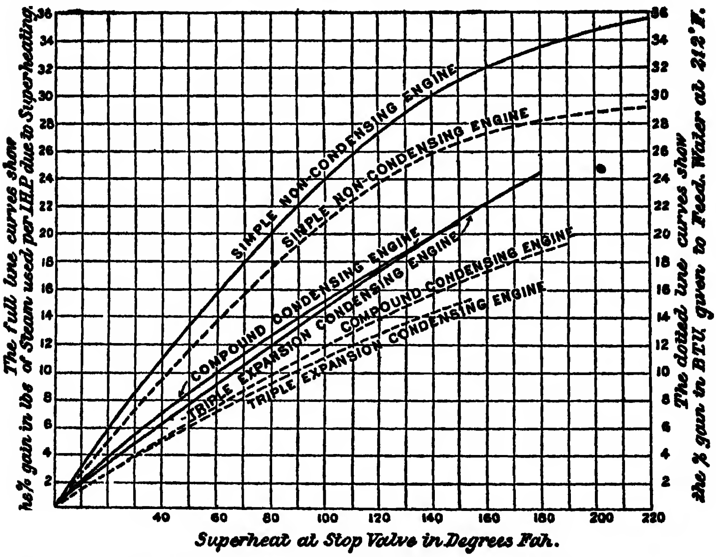


Fig. 17.—Curves Showing the Percentage Gain in B.T.U. given to Feed-Water, as well as the Percentage Gain in Steam used per I.H.P.-Hour Due to Superheating in Willans & Robinson's Simple, Compound, and Triple-Expansion Engines.

Consequently, since 20 lbs. of such steam was used, (20 × 1,093) = 21,860 B.T.U., as the total heat units in the superheated steam required per I.H.P.-hour, hence:—

$$\begin{array}{ccccccc} & \text{B.T.U.} & & \text{B.T.U.} & & & \\ & 30{,}707{\cdot}6 & : & 21{,}860 & :: & 100\% & : & y\%. \end{array}$$

$$\therefore \quad y = 71{\cdot}2\,\%.$$

Or, (100 per cent. – 71·2 per cent.) = 28·8 per cent., which is the net calculated gain when reckoned in B.T.U. added to feed-

water from 212° F. due to superheating, instead of the previously measured 35 per cent. gain in lbs. of steam used per I.H.P. In all cases, as shown by Fig. 17, it will be found that the difference between these two systems of estimating the gain due to superheating, increased with the superheat. When testing engines using superheated steam, it will be found interesting and instructive to plot down curves of their percentage gains by both methods.

The author has put the previous proportion sums into a simple formula for ascertaining the percentage gain in B.T.U. given to feed-water due to superheating in the following way.*

Let H_{su} = Heat units per lb. of superheated steam from temp. of feed-water to temp. of superheat.

„ H_{sa} = Heat units per lb. of saturated steam from temp. of feed-water to temp. due to pressure, p, in lbs. per square inch absolute at the steam chest.

„ W_{su} = Weight of superheated steam used per I.H.P.-hour at the stop-valve pressure, p, and temp. of superheat.

„ W_{sa} = Weight of saturated steam used per I.H.P.-hour at pressure p.

$$\left.\begin{array}{c}\text{Then, Percentage gain in B.T.U.}\\ \text{due to superheat}\end{array}\right\} = 100 - \left(\frac{100\,H_{su}\,.\,W_{su}}{H_{sa}\,.\,W_{sa}}\right).$$

But, $H_{sa} = (H - S)$ (see Lectures VII. and IX.).

Where H = Total heat in B.T.U. per lb. of feed-water from 32° F., as found from Table II. on "The Properties of Saturated Steam," up to and at pressure p.

And, S = Sensible heat in B.T.U. per lb. of feed-water from 32° F. to temp. of feed, t_f°. Or, $S = (t_f^\circ - 32^\circ)$.

Also, $H_{su} = H_{sa} + H_\sigma\, t_{su}^\circ$ (see Lectures IV., VII., and XI.).

Where H_σ = 0·48 the specific heat of steam and t_{su}° = superheat at steam chest in degrees Fah.

Substitute these values in the above formula :—

$$\text{Then, \% gain} = 100 - \left\{\frac{100\,[H - (t_f^\circ - 32^\circ) + H_\sigma\, t_{su}]\, W_{su}}{[H - (t_f^\circ - 32^\circ)]\, W_{sa}}\right\}.$$

Taking the same test and values as in the previous example for the simple non-condensing engine, where p = 80 lbs.; H = 1,177 B.T.U.; $t_f^\circ = 212^\circ$; $H_\sigma = \cdot 48$; $t_{su}^\circ = 200^\circ$; W_{su} = 20 lbs.; and W_{sa} = 30·8 lbs.

$$\text{Then, \% gain} = 100 - \left\{\frac{100\,[1{,}177 - (212 - 32) + \cdot 48 \times 200]\,20}{[1{,}177 - (212 - 32)]\,30\cdot 8}\right\}.$$

$$\text{Or, \% gain} = 100 - \left\{\frac{100\,[997 + 96]\,20}{997 \times 30\cdot 8}\right\} = 100 - 71\cdot 2 = 28\cdot 8.$$

* This formula was devised by Professor Jamieson for the discussion on Mr. F. J. Rowan's paper on "Superheated Steam." See *Proc. Inst. Engs. and Shipbuilders in Scotland*, vol. xlvii., February, 1904.

The gain in B.T.U. is therefore 28·8 per cent., as found before and from the test of Willans & Robinson's simple non-condensing engine, with a superheat of 200° F.

It will be seen, that the only variables in this simple formula are t_{su}° and W_{su}. Consequently, a constant can easily be found for the other values. The various calculations can, therefore, be quickly worked out for one complete set of trials at different degrees of superheat, their results marked on squared paper, a mean curve drawn through them, and comparisons made with tests of the same or of other engines for any agreed-upon temperature of the feed-water.*

The Effects of Raising the Superheat on the Indicator Card and on the Economy of Steam.—As an illustration of these effects, we reproduce the two mean indicator cards obtained by Prof. Ewing during his 1899 tests of the Schmidt superheater plant, to which we referred in the previous lecture when dealing with the "History of Superheating." The engine was a horizontal, single-acting one, with two side by side cylinders and the cranks at 180° apart. The pistons were 70·9 inches diameter, with a stroke of 11·8 inches, and a speed of about 175 revolutions per minute. The engine was made to work against a brake, and the B.H.P. was measured simultaneously with the I.H.P. The exhaust steam was collected in a surface condenser at atmospheric pressure, whilst the condensed water was weighed as well as the feed-water. The feed-water was 5 per cent. greater

* Since writing the foregoing and on going to press, I have just received the kind permission of The Institution to quote from "The (1898) Report of the Committee on the Thermal Efficiency of Steam Engines," appointed by The Institution of Civil Engineers, where they state :—

"(1) That the statement of the economy of a steam engine in terms of pounds of feed-water per I.H.P. per hour is undesirable.

"(2) That for all purposes, except those of a scientific nature, it is desirable to state the economy of a steam engine in terms of the thermal units required per I.H.P. per hour (or per minute), and that if possible the thermal units required per brake H.P. should also be given.

"(3) That for scientific purposes the thermal units that would be required by a perfect steam engine working under the same conditions as the actual engine should also be stated.

"The proposed method of statement is applicable to engines using superheated steam as well as to those using saturated steam, and the objection to the use of pounds of feed-water, which contain more or less thermal units according to conditions, is obviated, while there is no more practical difficulty in obtaining the thermal units per I.H.P. per hour than there is in arriving at the pounds of feed-water.

"For scientific purposes the difference in the thermal units per I.H.P. required by the perfect steam engine and by the actual engine shows the loss due to imperfections in the actual engine.

"A further great advantage of the proposal is that the ambiguous term 'efficiency' is not required."

DATA REFERRING TO FIG. 18 AND INDICATOR CARDS.

Data.	Low Degree of Superheat.	High Degree of Superheat.
Pressure of steam by gauge in lbs. per square inch at stop valve,	110	126
Temperature of steam close to engine,	494° F	640° F.
Amount of superheat in Fah degs. close to engine,	185° F.	320° F.
Revolutions per minute,	176	177
Load on brake in lbs.,	280	320
Brake horse-power,	15·54	18·33
Weight of steam in lbs. condensed per revolution,	0·0399	0·333
,, ,, ,, B.H.P.-hour,	26 5	19·4

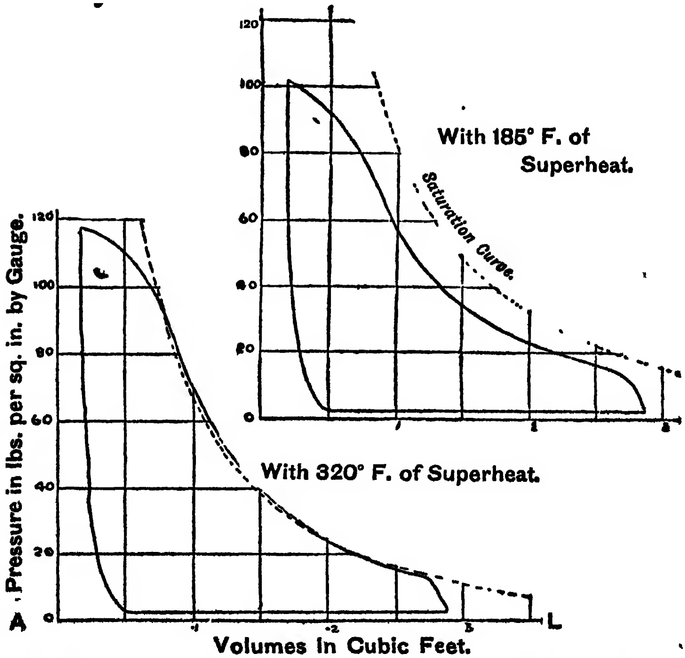


FIG. 18.—INDICATOR CARDS FROM THE SAME ENGINE WITH TWO DIFFERENT DEGREES OF SUPERHEAT.

than the exhaust steam collected in the condenser. This was probably due in part to the leakage of steam past the piston rings through the open ends of the cylinder.

In the upper diagram, the pressure between the stop valve and the inside of the cylinder fell from 110 to 102 lbs., whilst the steam in the cylinder is shown to be wet throughout, by the distance between the saturation curve and the expansion side of the card. Also, the steam contained about 24 per cent. of water at cut-off and about 18 per cent. of water just before release.

In the lower diagram, the pressure between the stop valve and the inside of the cylinder fell from 126 to 118 lbs., whilst the steam in the cylinder remained dry and almost coincides with the saturation curve throughout the expansion. It is very little superheated during the early stage of the expansion, whilst it becomes saturated shortly before release. It therefore appears, in this small open-ended, unjacketed engine, with a cut-off at about $\frac{1}{8}$ of the stroke, that the cooling action of the cylinder walls is such, that a superheat of 320° F. only suffices to make the steam dry at cut off and to keep it so during expansion. Whereas, a superheat of 185° F. does not prevent the steam from containing no less than 24 per cent. of water at cut-off, due to initial condensation and other causes.

This lesson, to engineering students, is a startling revelation, and shows most conclusively, that although the thermo-dynamic efficiency of *highly superheated steam* is relatively small, yet the benefits derived therefrom are chiefly threefold :—(1) The prevention of initial condensation; (2) the prevention of alternate condensation and re-evaporation in the cylinder; (3) that, with such leaky moving parts as trunk pistons, plain pistons, steam and exhaust valves without springs, the leakage of steam past these sliding surfaces is much less with highly superheated steam than with medium superheats, or with dry saturated steam. The B.H.P. increased nearly 18 per cent. with far less than this increase in mean steam pressure. This is, however, not the best result which Professor Ewing obtained from an engine using highly superheated steam on the Schmidt system, for in 1903 he got the remarkable figure of 9 lbs. of steam per I.H.P.-hour.

APPENDIX TO LECTURE XVI.

The Planimeter.*—It is frequently necessary for engineers to ascertain the areas, and mean lengths or breadths of irregular flat figures, such as plans of properties, countries, ships, and diagrams of work done by engines, dynamos, and other machines. In order to explain how such areas and mean heights may be obtained by aid of this instrument, we shall first of all describe the construction and action of Amsler's Planimeter and the method of reading its scales.

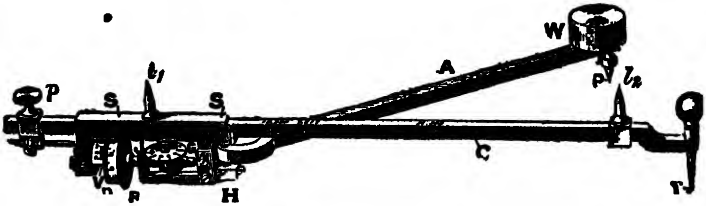


FIG. 1.—AMSLER'S PLANIMETER AND ITS RECORDING MECHANISM.

Amsler's Planimeter.—From Fig. 1, it will be seen that this instrument consists of two metallic arms or bars, A and C, hinged together at H on the sleeve, S. One arm, A, carries, at its free end, a weight, W, and a needle point, P, which acts as a fixed pivot for the instrument. The other arm, C, carries a tracing point, T, with which the outline of any figure to be measured is traced, whilst the third supporting point consists of the roller, R, on which the whole freely moves over the diagram.

By referring to the enlarged Fig. 2, the details of the counting mechanism will be better understood. The roller, R, carries a drum, D, which is graduated to record the *area* traced by the point, T. There is also a set pin, *p* (Fig. 1), with adjusting nut and screw, by which the arm, C, may be fixed to the sleeve, S, at any desired position to give a convenient scale.

* I am indebted to the Crosby Steam Gage and Valve Company's *American Indicator Pocket-Book* for four of the first five figures in this article. There are many kinds of planimeters or integrators of areas, as will be seen from a perusal of Prof. Hele Shaw's paper on "Mechanical Integrators," read before the Inst.C.E. (see *Proc.*, vol. lxxxii., paper No. 2,063). I have chosen the Polar Planimeter, invented by Prof. Amsler-Laffon, for description here, because it is the one now most commonly used by engineers for ascertaining the areas and mean pressures of engine indicator diagrams.

Recording Mechanism.—The *second* figure shows in detail the recording mechanism of the planimeter. The drum, D, of the roller-wheel, R, is divided into 10 equal and larger parts numbered 1 to 10. Each part or number represents *one* square inch for a certain position of C in S as shown at K. The distances between each of these 10 numbers are subdivided into 10 equal parts, each one of which represents *one-tenth* of a square inch. The vernier, V, has 10 divisions, each of which is one-tenth less than any of the 100 on D. Consequently, if one division on V exactly coincides with another on D, the distance—counted from zero—represents so many *hundredths* of a square inch.

The graduated wheel, G, is geared to the roller, R, in such a manner as to rotate once for *ten* revolutions of R. The face of this wheel is divided by radial lines into *ten* equal and numbered parts, each one of which represents *ten* square inches. It therefore indicates the revolutions of the roller wheel, R.

Method of Reading the Scales.—Now, supposing that a certain area has been measured from zero, the result may be read off as follows:—

(1) Find the numbered radial line on G (Fig. 2) which has just passed the mark line on the fixed arm, J. Say it is 1. This represents *ten.*

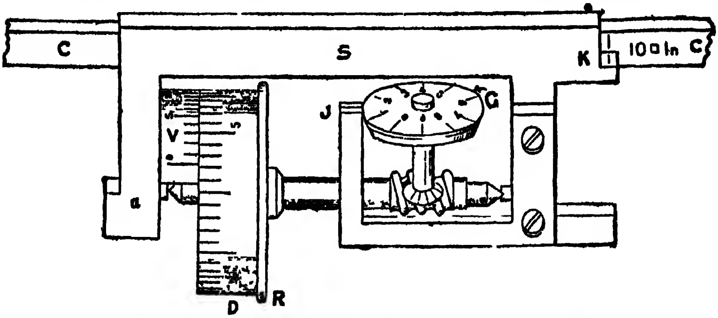


FIG. 2.—RECORDING MECHANISM OF AMSLER'S PLANIMETER.

(2) Find the number on the drum, D, which has passed zero (or 0) on the vernier, V. Let this be 4. This represents 4 *units.*

(3) Further, let the number of subdivisions over 4 be 7, as shown by the dotted line, *a*. This 7 therefore represents $\frac{7}{10}$ or ·7.

(4) Find the graduation number on the vernier, V, which exactly coincides with a division line on D. Let this be the *third* one from zero. It therefore represents 3 *hundredths.*

Then, as a whole, we have 14·73 square inches as the complete reading which represents the full area of the figure that has been traced in outline by the point, T.

If the movement of the roller wheel, R, had been 3 one-hundredths *less*, its seventh graduation would have coincided with the zero of the vernier, V, and the reading would then have been 14·70 instead of 14·73.

Directions for Using Amsler's Planimeter.—This planimeter is a precise and delicate instrument. It should be handled and kept with great care in order that it may be depended upon to give accurate results.

It is also necessary to have a *flat, even, unglazed surface for the roller wheel,* R, *to travel upon.* A piece of dull finished cardboard will serve the purpose very well.

To find the Area of a Figure with the Planimeter.—(1) Place the instrument on the drawing in the position shown graphically by Fig. 3, so as to allow perfect freedom of motion in every direction in which it requires to move. Place the weight at P, and press the needle-point down gently, so that it will just stick to that place.

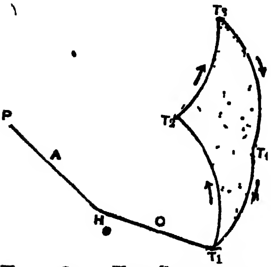


FIG. 3. — TO SHOW HOW AREAS ARE TRACED AND FOUND BY THE PLANIMETER.

(2) Put the point of the tracer, T, upon any given point in the outline of the figure. Do not waste time in attempting to set the scales to zero, but take the *initial* reading, as previously directed, wherever they happen to stand. Follow the outline of the figure carefully with the tracer-point, T, by moving it in the direction indicated by the arrows until it returns to the starting point. That is, move from T_1 to T_2 to T_3 to T_4 back to T_1. Then the scales must be read off for the *final* reading, and the difference of the two gives the area, *provided* P *lies outside the figure.*

N.B.—Great care must be taken to have the instrument *in its proper position* for tracing the outline of the figure before taking the *initial* reading. Also, the *final* reading should be taken as soon as the tracing is completed, because the least movement of T will change the result.

To measure Indicator Diagrams with the Planimeter.

1. *To obtain the Area in Square Inches.*—By referring to Figs. 1 and 2, it will be observed that there are vertical numbered marks on the front side of the bar, C. Now, when set pin, *p*, is slackened, the bar, C, may be pulled out or pushed through the sleeve, S, until the line mark, K, on the right hand of S is opposite to a vertical numbered mark on the side of C. For example, if we desired to measure the area of the diagram in square inches, the line at K should be brought fairly opposite the line marked 10 square inches on C; because, one complete revolution of the roller, R (which is 2·5 inches in circumference), will indicate 10 divisions or 10 square inches on its drum, D. Then tighten the set-pin, *p*. The exact distance between the tracing point, T, and the hinge, H, will now be 4 inches—*i.e.*, the radial length of the arm, C, is 4 inches, and the distance between the two upper pointed pins, l_1 to l_2, will also be 4 inches.

Now, by running clockwise round the diagram with T (as indicated by the arrows in Fig. 4, in the manner previously described), the difference between the *initial* and *final* scale readings will indicate the area of the diagram in square inches—

For, (Length of arm, C) × (Circumference of R) = Area of diagram.
4″ × 2·5″ = 10□ inches.

The accuracy of the instrument to indicate square inches, may be tested by drawing exact fine line squares of 1, 2, or 3 inches sides, and passing T carefully along them clockwise. When straight lines have to be followed, a thin straight rule may be placed close alongside of them to guide T.

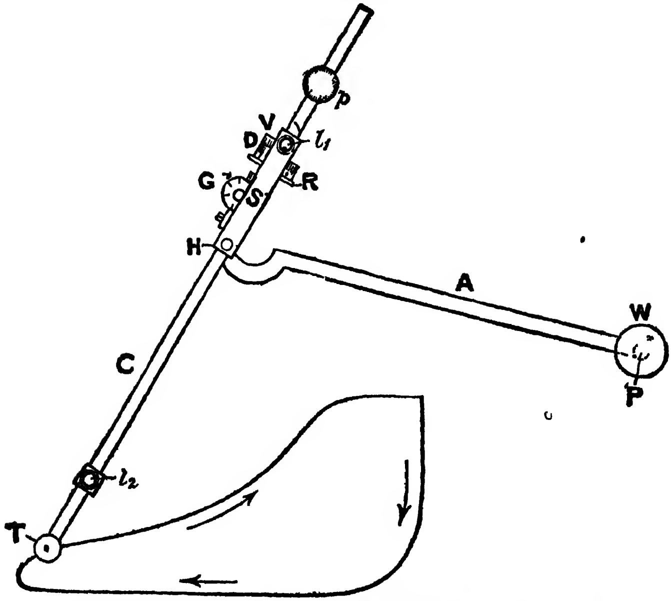


FIG. 4.—FINDING THE AREA OF AN INDICATOR DIAGRAM.

2 *To obtain the Average Height or Mean Pressure of a Diagram.*—(*a*) Slacken the set pin, *p*, and push in or pull out the bar, C, until the two steel points, l_1 (on the upper side of the sleeve, S) and l_2 (on the upper side of the bar, C), exactly divide off the length of the diagram between them along the atmospheric line, or parallel to it, as shown by Fig. 5. The

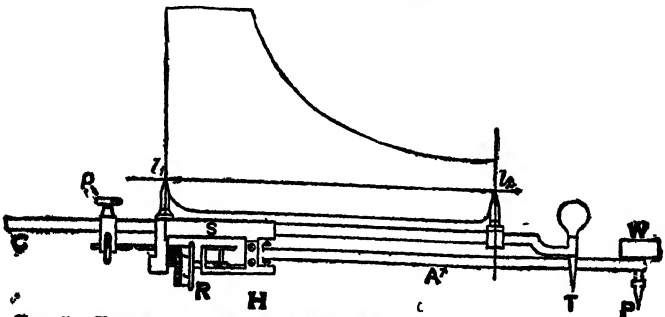


FIG. 5.—FINDING THE MEAN PRESSURE FROM AN INDICATOR DIAGRAM.

final adjustment is made by the little nut and screw seen under the pin, p, in Figs. 1 and 5, *after* the set pin is tightened. With this adjustment, the figures on the counting disc, G, represent *hundreds*, those on the roller disc, D, *tens*, and each of its finer 100 divisions *units*, whilst the vernier, V, will give the decimals.

(b) Place the instrument in the position shown by Fig. 4, and trace the outline of the diagram as previously directed. The difference of the readings will be its *average height* in fortieths of an inch.*

$$\text{Since, } \left.\begin{matrix}\text{(Average height of}\\ \text{diagram)}\end{matrix}\right\} \times \text{(Length of diagram)} = \text{Area of diagram.}$$

$$\text{And, } \left.\begin{matrix}\text{(Net motion of}\\ \text{roller, R)}\end{matrix}\right\} \times (\text{Distance, } l_1 \text{ to } l_2) = \text{,,}$$

$$\therefore \quad \text{(Net reading on D)} \times \text{(Arm, C, or H T)} = \text{,,}$$

Since the distance between l_1 and l_2 is always equal to length of arm, C, or H T.

$$\therefore \quad \textit{Average height of diagram} = \textit{Net reading on } \text{D}.$$

Suppose, that after measuring the diagram, we read from the figures on the roller disc, D, and its intermediate divisions, and from the vernier, V, also, 3, 5, and 2 respectively. Then, we have 35·2 fortieths of an inch, which, divided by 40, gives ·88 of an inch as the average height. This, multiplied by the scale of the spring used (which in this case we assume to be 60 lbs per lineal inch), gives 52 8 lbs. as the *mean effective pressure* per square inch on the engine piston area. A simple method is to multiply the reading by the *factor* corresponding with the scale of the spring, which, for a 60-lb. spring, is $(60 \div 40) = 1{\cdot}5$.

Or, *Mean pressure per square inch* = 1·5 (mean height of diagram).

3. *To obtain the Mean Effective Pressure of Looped Diagrams.*—When taking indicator cards of engines, instances occur where the back pressure line rises above the forward pressure line, due to excessive compression. Then, part of the indicator diagram is positive while the other part is negative, as shown in Fig. 6, by the hatched and unhatched portions respectively. Consequently, we must have the area of the unhatched portion deducted from that of the hatched portion when the mean effective pressure is calculated. In order that the planimeter should effect this deduction automatically, the tracer-point, T, should be caused, in traversing the loops and lines, to move upon every portion of them in the same direction as that in which they were drawn upon the paper by the indicator-pencil.

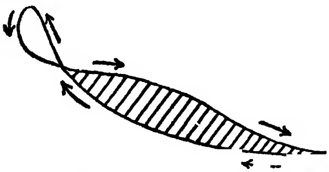


FIG. 6.—FINDING THE MEAN EFFECTIVE PRESSURE FROM A LOOPED DIAGRAM.

* Since the roller, R, is 2·5 inches in circumference, and its scale on drum, D, is divided into (10 × 10) 100 equal parts, each of these fine divisions must represent $(2{\cdot}5 \div 100) = \frac{1}{40}$ inch.

Mathematical Explanation of Amsler's Planimeter.—The simplest and clearest mathematical explanation of Amsler's Planimeter which I have seen, is to be found in *The Philosophical Magazine*, vol. xlviii., Fourth Series, by F. P. Purvis.* I have altered his index letters to correspond with those indicating the same parts in the previous figures and added Fig. 10 to illustrate the latter part of his explanation.

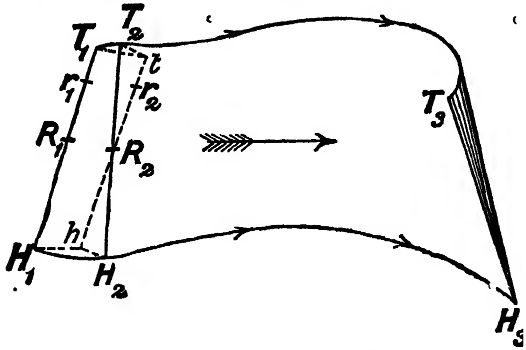


Fig. 7.—To Find the Travel of a Simple Bar, $T_1 H_1$.

Suppose that the instrument consisted simply of the straight bar, T_1H_1, of length, l, carrying a pencil at each end, T_1 and H_1; and, suppose any lines, T_1T_2 and H_1H_2, to be traced out by these two pencils. Then, let us consider how the area, $T_1T_2H_2H_1$, may be expressed in terms of the length, l, of this bar, and the motion of some point in the same.

Let the motion from T_1H_1 to T_2H_2 represent an elementary motion of the bar, the centre of it R_1, moving from R_1 to R_2, and the bar turning about R_2 through the angle $(d\theta)$. Let (dn) be the normal distance from R_2 to T_1H_1. This motion may be considered to take place in two parts:—

1st, the motion of T_1H_1 parallel to itself into the position, th.

2nd, the motion of T_1H_1 when at th about R_2 into the position, T_2H_2. The required area, $T_1T_2H_2H_1$ (in this elementary motion), is equal to the area, T_1thH_1. But this area is also equal to $l\,(dn)$, since the area, R_2T_2t = the area, R_2H_2h, and the areas, T_1T_2t and H_1H_2h, are negligible with respect to $l\,(dn)$, being the product of two infinitesimal quantities, while $l(dn)$ is the product of one infinitesimal quantity (comparable with each of the two just mentioned) and the finite quantity, l.

Integrating for the whole area, $T_1T_3H_3H_1$, we see, that it is expressed by $(l \times n)$, where n is the travel of the point, R_1, normally to the bar, T_1H_1.

Now, we may obtain that normal motion, n, by centring a wheel on the bar at R_1, free to revolve in the plane at right angles to T_1H_1, and resting at its circumference on the paper. That, n, is given by the circumferential motion of this wheel, may be seen by again considering the elementary

* Prof. Purvis was Senior Whitworth Scholar in the first competition of 1869, and he is at present Professor of Naval Architecture at the Imperial University, Tokio, Japan.

motion of the bar from $T_1 H_1$ to $T_3 H_3$. While the bar moves from $T_1 H_1$ to th, the wheel turns through the normal distance from R_2 to $T_1 H_1$. While the bar turns about the point, R_2, the wheel remains stationary.

If, instead of centring the wheel at R_1, we centre it at any other point, say r_1, which may be at a distance, m, from R_1, then its circumferential travel for the elementary motion will be the normal length from r_2 to $T_1 H_1$, or $(dn) - m(d\theta)$ And, for the whole motion from $T_1 H_1$ to $T_3 H_3$, the travel will be $(n - m\theta)$, where θ = the inclination of $T_3 H_3$ to $T_1 H_1$.

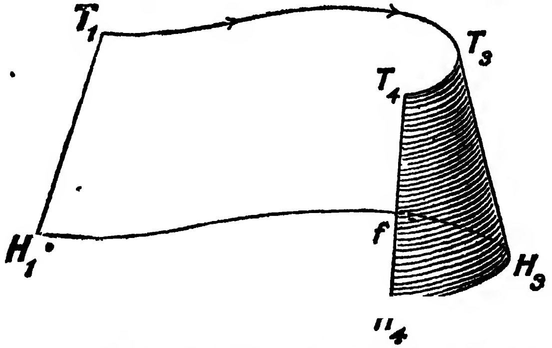


FIG. 8.—SHOWING RETROGRADE MOTION OF THE BAR FROM $T_3 H_3$ TO T_4 H

If a retrograde motion be now given to the instrument, bringing it int the position, $T_4 H_4$, the product $(l \times n)$ will still equal the area include between the two curved lines ($T_1T_3T_4$ and $H_1 H_3 H_4$) and the two straigh lines ($T_1 H_1$ and $T_4 H_4$) Part of this area is shown negative, o $(l \times n) = (T_1T_3T_4 f H_1 - f H_3 H_4)$ If, instead of allowing H_1 to take an path, $H_3 H_4$, we constrain it to move only along the line already traced, whil T_1 traces out a new line, T_3T_4, then the negative area will be nil and th

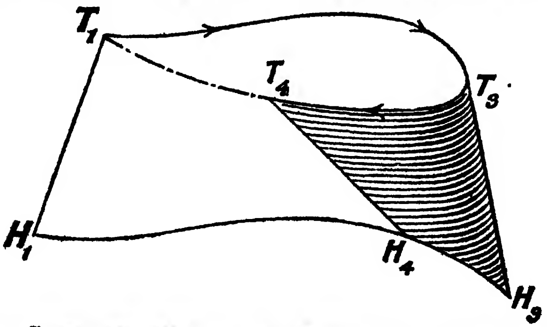


FIG. 9.—CONSTRAINED MOTION OF ONE END, H_1, OF THE BAR, $T_1 H_1$.

product $(l \times n)$ will equal the area, $T_1T_3T_4 H_4 H_1$. If this motion be continued, H_1 being always kept in the path, $H_3 H_4 H_1$, until $T_1 H_1$ occupies

its initial position, the product ($l \times n$) will equal the area, $T_1 T_3 T_4 T_1$, whatever be the nature of the line, $H_1 H_4 H_3$. Also, for the whole motion $\theta = 0$, so that the circumferential travel of the wheel at r_1 is equal to n, and is entirely independent of the value of m. Now, in Amsler's planimeter, the point, H_1, is constrained to move in the arc of a circle, while the pencil, T_1, is traced round the contour of the required area. This is simply a limitation of the more general and previous case, and, it is clearly shown by Fig. 10, where the points, H_1, H_2, H_3, &c., move to and fro along the arc of the circle whose radius is $P H_1$, whilst the tracing point, T, describes the figure, $T_1 T_2 T_3 T_4$, back to T_1.

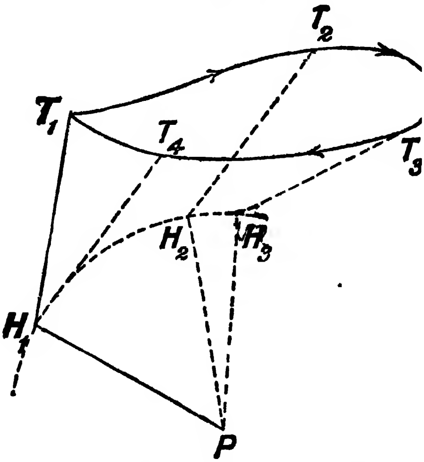


FIG. 10.—END, H_1, OF ARM, $T_1 H_1$, IS CONSTRAINED TO MOVE ALONG THE ARC OF A CIRCLE.

Also, the wheel whose travel is measured, is placed away from the centre of the bar; in fact, on the opposite side of H_1, but, as we have seen, its position, so long as its centre is on the line, $T_1 H_1$, is quite immaterial, for its motion in the aggregate is the same as if it were placed at R_1.

In the planimeter the length, l, is capable of variation. By setting it differently, the same graduation on the wheel will give areas in different units, the unit of area being always $l \times$ the circumferential travel of the wheel required to alter its reading by unity.

Those who are anxious to now study the mathematical proofs, as well as popular explanations of this and other kinds of integrators may consult :—

(1) *British Association Reports*, 1872, p. 401, for Sir Frederic Bramwell's paper.

(2) *The Philosophical Magazine*, vol. xlviii., 4th series, July, 1874, for a paper by Sir F. P. Purvis, Wh.Sc., &c., now at Japan University.

(3) *Proc. Inst. C.E.*, vol. lxxxiii., 1885, for paper on "Mechanical Integrators," by Prof. Hele Shaw, of Liverpool University.

(4) *Proc. Inst. Junior Engineers*, Dec., 1902, for Mr. W. J. Tennant's paper on "The Planimeter Explained without Mathematics."

LECTURE XVI.—QUESTIONS.

1. Sketch in section and plan Watt's original indicator. Describe the instrument and show how it was applied to obtain the mean pressure in a steam cylinder.

2. Sketch and describe concisely the Crosby indicator. Point out how and why it differs from other indicators with which you are acquainted.

3. Draw the normal indicator diagram of a condensing engine, and trace the changes in outline produced by the principal causes which may, in practice, detract from the efficiency of the engine.

4. Describe and sketch any one form of steam engine indicator with which you are acquainted. Why are modern indicators made more rapid in their natural vibrations and what means are taken to effect this object? What sort of errors do we expect to find when an engine-driver takes diagrams, and what are they due to? (S. & A., 1897, Adv.)

5. The barrel of an indicator is 2 inches in diameter, and it vibrates through $\frac{3}{4}$ of a revolution. The area of the diagram is $3\frac{3}{8}$ square inches, and the motion of the pencil is 3 times that of the indicator piston. Taking the mean pressure of steam to be $17\frac{1}{2}$ lbs. per square inch, find what force corresponds to a motion of 1 inch of the spring. *Ans.* 67·5 lbs. nearly.

6. Sketch and describe an indicator for an engine of 250 revolutions per minute. Why are its requirements different from those for an engine running at 80 revolutions per minute? Show how it is connected up and how a diagram is taken. (S. & A., 1898, H., Part i.)

7. Enumerate the several errors to which indicators are liable. Why should the inertia of the moving parts of an indicator be a minimum?

8. Sketch and describe the several methods of obtaining the reduced motion of the piston when taking indicator cards. State which you consider to be the best arrangements for giving good results, and why?

9. Give a clear, concise description of how you would take the indicator diagram of an engine, giving the necessary sketches to illustrate your answer. How would you attach the indicator to the engine cylinder, and why?

10. A non-condensing engine is using steam at 42 lbs. per square inch above the atmosphere—the length of the stroke is 3 feet, and steam is cut off at $\frac{1}{3}$ stroke—draw an approximate diagram (scale $\frac{1}{24}$) marking points of release and compression, and showing the direction of motion of the piston by arrows. Find, by calculation, the mean pressure. *Ans.* 24·9 lbs.

11. Draw indicator diagrams as commonly given in a double-acting engine, (1) of the condensing type, (2) when non-condensing.

12. Draw the ordinary indicator diagrams as obtained (1) from the top, (2) from the bottom of the cylinder of a single-acting condensing engine, and account for the difference in form of the respective diagrams.

13. Show effects of wire-drawing and of clearance upon an indicator diagram. What is the object of a steam-jacket? In what way does the absence of the jacket affect the indicator diagram?

14. Show by sketches and explain the effects on an indicator diagram of (1) deficiency of lead; (2) deficiency of outside lap; (3) contracted long steam passages; (4) initial condensation; (5) leaky admission valves; (6) leaky piston; (7) too much inside lap; (8) leaky condenser.

15. Explain and indicate on separate diagrams, by comparison with the normal indicator diagram, the effect of (1) wire drawing on the admission of steam, (2) wire-drawing on the exhaust side, (3) a leaky slide-valve,

(4) a leaky piston, (5) the cushion pressure exceeding the pressure of the initial steam. In the last case, how is the area of the diagram calculated? Further, if the dimensions of the parts and proportions of the slide-valve are correct, but (*a*) the fixing of the valve on the valve-spindle is incorrect, (*b*) the angle of advance of the eccentric is too small, and (*c*) the angle of advance of the eccentric is too great, what would be the effect separately of (*a*), (*b*), (*c*) on the working of the engine, and show their effect on the indicator diagram. (S. & A., 1897, Hons.)

16. Suppose you took an indicator diagram from a high speed engine going at 400 revolutions per minute and found the admission and the expansion line to be an up and down wavy line like Fig. 8 in this lecture. To what would you attribute this defect, and what would you do in order to obtain a smooth firm outline?

17. Explain the operation of combining the indicator diagrams of work done in a compound cylinder engine, the object being to produce the diagram which would have been obtained if the steam had performed the same work by going through the same changes of pressure and volume in one cylinder.

18. A compound condensing engine with cranks at right angles has cylinders of 20 inches and 35 inches diameter with 3 feet stroke. The high-pressure cylinder has a clearance of $\frac{1}{16}$ and the low-pressure one of $\frac{1}{12}$ of the volume of their respective cylinders Dry saturated steam of 100 lbs. absolute is admitted to the high-pressure cylinder and is cut off at $\frac{1}{4}$ stroke, whilst the cut-off in the low-pressure cylinder is at $\frac{1}{2}$ stroke. Let both cylinders be well jacketed and the vacuum 28 inches. Draw the probable indicator diagrams and find the mean pressure in each cylinder. Plot down a combined diagram with the probable correct position of the saturation curve and attach a scale of pressures and volumes to your figure.

19. In a single acting engine it is necessary to take one indicator diagram from above and another from below the piston. Sketch each diagram in juxtaposition so as to form a single compound diagram, and explain generally the reasons for the different outlines of the diagrams. To what cause do you attribute the space between the diagrams?

20. How would you ascertain from an indicator diagram the probable percentage of condensed steam at cut-off and during expansion?

21. State and indicate, by a scale diagram with an example, why it is preferable to estimate the per cent. gain in B.T.U. per I.H.P rather than the per cent. gain in feed-water or steam used per I.H.P. Analyse Jamieson's formula given in this lecture, and state the conclusions of the Inst. C. E. Committee.

22. Show by two diagrams the effects of supplying a simple non-condensing engine (1) with dry saturated steam; (2) with highly superheated steam. State whereby the economy chiefly arises in the latter case.

23. What is a planimeter, and what are its uses? Sketch an Amsler's planimeter and give a descriptive index of its parts showing how it is used to ascertain the area and the mean pressure of an indicator diagram.

24. How would you use an Amsler planimeter to find the mean pressure of a looped diagram?

25. Give a concise, clear explanation with figures of how Amsler's planimeter measures the area of a diagram of work.

26. Sketch an indicator diagram such as might be expected from a non-condensing engine with a slide-valve. If the weight of water present during cushioning is known, and the feed water per hour is also known, show how we find how much condensation or evaporation occurs during the expansion. (B. of E., 1900, Adv.)

27. Describe the construction of an indicator and how it is used. Give a sketch of a specimen indicator diagram from a steam, gas, or oil engine, and describe what each part means. What sort of information is given to us by an indicator diagram? (B. of E., 1903, Adv.)

28. Sketch a typical indicator diagram for an engine in which the cut-off takes place at one-third the stroke by a single slide-valve worked by an eccentric. Make a sketch of the section of the cylinder showing the valve in its middle position, and sketch also the connections between the valve and crank axle. (C. & G., 1903, O., Sec. C.)

29. State what data you require in order to construct a mean diagram showing the amount of steam missing at any point of the stroke of a steam engine. Point out precisely what assumptions you make, and sketch the diagram. (C. & G., 1903, H., Sec. B.)

30. A compound condensing engine, with cranks at right angles and an intermediate receiver, has cylinders of 14 and 24 inches diameter respectively, each with a stroke of 36 inches. Draw the indicator diagrams which you would expect to obtain from the cylinders supposing steam of 90 lbs. absolute pressure is admitted to the high-pressure cylinder and is cut off at half-stroke, the steam in the low-pressure cylinder being cut off at $\frac{5}{8}$ stroke, and the condenser showing a back pressure of 4 lbs. absolute. Attach a scale of inches and pounds to your diagram.

31. Suppose that you started with an old "steam eater" of an engine of the *non-condensing*, *unlagged* type, using ordinary saturated steam of 20 lbs. absolute, running at 50 revolutions per minute, cutting off at $\frac{3}{4}$ stroke, with a clearance of $\frac{1}{8}$ of piston's displacement; and found, by indicator cards, that it was using 100 lbs. of steam per I.H.P.-hour. If you applied to this engine the following improvements in succession, you would probably find the consumption of steam reduced by about the stated amounts. Calculate the percentage diminution in steam which each improvement makes upon the previous case, and as a whole upon the engine. State clearly the several percentage savings in the form of a table. Plot out curves with percentages as ordinates, and lbs. of steam used per I.H.P.-hour as abscissæ. Mention any omitted improvements and their percentage values.

		Lbs. of Steam per I.H P.-hour.
(1) *Lag pipes, cylinder* and valve casing, . .	when you use	80
(2) *Increase pressure* to 65 lbs. absolute and cut-off $\frac{1}{4}$ stroke, ...	,,	75
(3) *Increased speed* to 100 revs. per minute, .	,,	70
(4) *Reduced clearance* to $\frac{1}{20}$ of piston's displacement, employing Corliss or drop-valve gear,	,,	60
(5) *Condensing.*—Next adopt condensing with 2 lbs. back pressure, ...	,,	40
(6) *Steam Jacketing.*—Next steam jacket the engine thoroughly, ...	,,	30
(7) *Compounding.*—Next increase the initial pressure to 100 lbs. and adopt compound condensing to the best advantage, . .	,,	25
(8) *Increased speed.*—Increase the *r.p.m.* to 300,	,,	20
(9) *Triple expansion.*—Increase steam pressure to 150 lbs per square inch, and use triple expansion, ...	,,	15
(10) *Superheating.*—Superheat the steam of the previous case by 150° F., and use re-heating,	,,	12

LECTURE XVII.

CONTENTS.—Nominal and Indicated Horse Power—Rule for finding the Indicated Horse Power of an Engine—Formula for finding the Mean Pressure—Brake Horse Power—Prony Brake or Absorption Dynamometer—Society of Arts Rope Dynamometer—Advantages of the Rope Brake—Tests of Small Engines with the Rope Brake—Questions.

Horse-Power.—The unit of power which is universally adopted by mechanical engineers in this country is that which was proposed and used by Watt—viz., *the horse-power.*

The steam engines introduced by Watt, were employed to a large extent in doing work which had formerly been done by horses, and hence it became necessary for him to be able to state the number of horses to which his engine would be equivalent in power. Watt estimated the power of the strongest London horses as about equal to that required to raise 33,000 *lbs. one foot high in one minute,* and he adopted this as his standard of power. This estimate, however, is too large, the average power of a horse being only about 22,000 foot-pounds* per minute, but Watt seems to have been desirous that his engines should exceed, rather than fall short of, their nominal power.

What is, therefore, technically spoken of among engineers as a *horse-power*, is the rate of doing work corresponding to 33,000 foot-pounds per minute, and the power of steam engines is always calculated on this basis.

Watt found that in his engines, he usually obtained a mean pressure of about 7 lbs. per square inch in the cylinder, and he estimated the power of his engines by assuming that value for the mean pressure. The horse-power thus estimated, he termed the *nominal* horse-power, and in practice that power was actually obtained. When, however, increased steam pressures came into general use, the mean pressure of steam in the cylinders could no longer be correctly taken as 7 lbs., and the nominal horse-power differed largely from the actual horse-power. In commerce the term nominal horse-power had been so much used, that commercial men understood the size, and, therefore the value, of an engine much better when its nominal horse-power was spoken of than its actual power, and, therefore, the term

* The foot-pound is the unit of work, and is the work done by a force of one pound acting through the space of one foot.

was retained for a long time, and even yet is still used for some classes of engines, such as those used for agricultural purposes. However, as unfair competition often takes place between different manufacturers, owing to the use of this term, it is fast falling into disuse and should be altogether abandoned.

The actual power exerted in the cylinder of an engine, cannot be obtained until we know the actual mean pressure of steam in the cylinder. In order to ascertain this, we must take a diagram from the cylinder by means of the indicator which was described in the last Lecture. The horse-power obtained by this means is termed the *indicated* horse-power, and when the horse-power of engines is spoken of, it is the indicated horse-power (I.H.P.) which is understood unless otherwise stated.

The diagram at p. 229 is taken from a horizontal non-condensing engine, and from it we wish to find the mean pressure of steam in the cylinder. To do this, divide the diagram into ten equal parts, by aid of the parallel ruler accompanying the indicator, then read off the pressures at the *centre* of each space or division, as described at p 151, and shown by the vertical lines in Fig. p. 229, by means of the scale corresponding to the indicator spring. The sum of these pressures divided by 10 gives the mean pressure during one stroke. This is shown worked out on the diagram, the mean pressure in this case being 43·5 lbs. per square inch. Now the work in foot-pounds done by an engine in one minute is = total mean pressure on the piston in lbs. × distance in feet travelled by piston in one minute. But one horse-power is equal to 33,000 foot-pounds per minute.

Therefore, the horse-power exerted by an engine is = *total mean pressure on the piston in lbs.* × *distance in feet travelled by the piston in one minute* ÷ 33,000.

Let p denote the mean pressure of steam in lbs. per square inch.
" A " the area of the cylinder in square inches.*
" L " the length of the stroke in feet.
" N " the number of strokes per minute = revolutions × 2.
" H P " the horse-power.

Then, total mean pressure on the piston in lbs. = $A p$, also, distance in feet travelled by piston in one minute = $L N$.

$$\therefore \text{ the horse-power of the engine } = \frac{A p L N}{33,000}.$$

This formula is easily remembered, since it may be written so as to form the word "PLAN," *thus:—Horse-power* = PLAN ÷ 33,000.

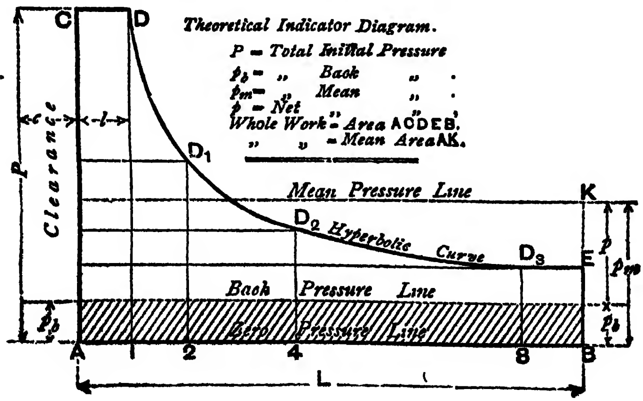
* In all cases, the area of the piston-rod has to be taken into account. For example, where the piston-rod comes out at the crank end of the cylinder only, then, A, *should be total area of cylinder less half the area of the piston-rod.*

Applying our formula to find the horse-power of the engine from the diagram and data given in it, we get

$$\text{H P} = \frac{\text{A}p\,\text{L N}}{33{,}000} = \frac{153{\cdot}9 \times 43{\cdot}5 \times 2 \times 80}{33{,}000} = 32{\cdot}45.$$

The diagram only gives the mean pressure on one side of the piston; but in practice it is usual to take the mean of two diagrams—one taken from each end of the cylinder. If there be two or more cylinders, the power developed in each has to be added together, in order to obtain the total horse-power.

If the student refers to questions 16, 17, 18, and 19 at the end of this Lecture, he will observe that he is given in each case the pressure of the steam on admission to the cylinder, the position of cut-off, *and the hyperbolic or napierian logarithm of the ratio of expansion*, as well as the diameter or the area of the cylinder, and the length of the stroke; from which, he is expected to calculate the net or effective work done in one stroke, or else the indicated horse-power when the number of revolutions per minute is stated. Now, as this is a very common form of question set in examination papers, and as the solution thereof will aid us in still further explaining (what has already been referred to at the end of Lecture XII.), that the area of the calculated or of the actual indicator diagram is a measure of the work done in one stroke, we shall first of all show how the hyperbolic logarithm is to be applied, in order to ascertain the mean total pressure throughout the stroke, *on the assumption that the steam expands according to Boyle's Law, neglecting clearance*, and secondly, we shall take into account the effect of clearance.



Referring to the last theoretical indicator diagram, A C D E B, the area of the rectangle, A D, is the product of the pressure line, A C, and the volume line, C D, or, A 1, to the point of cut-off, and therefore this area, A D, expresses the *whole* work done upon the piston by the steam in entering and in occupying that part of the cylinder before cut-off takes place; further, since the steam is supposed to expand in accordance with Boyle's Law ($p v =$ a constant), the curved line, D E, is a hyperbolic or isothermal curve, and the hyperbolic area, 1 D E B, expresses the *whole* work done by the steam during expansion—*i.e.*, after cut-off takes place. This latter area, 1 D E B, and consequently the *whole* work done during expansion, may be calculated by taking advantage of the known relations of hyperbolic curve areas to their base lines.*

For, if the base lines

A 1,	A 2,	A 4,	A 8,	&c.,

increase in the following geometrical progression,

as 1, to 2, to 4, to 8, to &c.,

then the successive areas,

—	$1\,D_1$,	$1\,D_2$,	$1\,D_3$,	&c.,

increase in the following arithmetical progression,

as — to 1, to 2, to 3, to &c.

For example:—

Let the area or volume, A D, up to the point of cut-off, be expressed by, 1, and the areas or volumes due to the expansion of the steam by the following numbers in geometrical progression:—

1, to 2, to 4, to 8, to &c.

* On the principle of logarithms, which represent in arithmetical progression natural numbers in geometrical progression, tables of hyperbolic logarithms are compiled to facilitate the calculation of the areas of work done due to various degrees of expansion. The hyperbolic logarithms are specially indicated or distinguished from common logarithms in formulæ by the small Greek letter ϵ, thus $\log._\epsilon$, and a few of these hyperbolic logarithms have been selected and printed (see Index), in order to enable students to work any of the ordinary questions. Hyperbolic numbers consist of the multiples of common logarithms by 2·302585, which, thus modified, become direct expressions of the actual ratio of the *whole* work done during expansion (due to different degrees of expansion) to the *whole* work done by the steam before expansion takes place.

The hyperbolic logarithms of these numbers are (see table, p. 269)

	·000,		·693,		1·386,		2·079,		&c.,
being as	0,	to	1,	to	2,	to	3,	to	&c.,

or in arithmetical progression; therefore, the *whole* work done by a quantity of steam expanded successively from the initial volume, 1,

being as	1,	to	2,	to	4,	to	8,	to	&c.,

will be in the proportions of

1, to 1 + ·693 to 1 + 1·386 to 1 + 2·079 to &c.,

or as 1, to 1·693 to 2·386 to 3·079 to &c.

Or generally if, r, be the ratio of expansion the whole work done will be as $(1 + \log_e r)$, showing that for an expansion of eight times, the initial work done by the steam before cut-off takes place, is tripled for that number of expansions by the end of the stroke. It is necessary, however, to deduct the work spent against the back pressure (due to an imperfect vacuum reckoned from the absolute zero or perfect vacuum line), before we obtain the net or effective work done by the steam in one stroke.

Another method of reasoning out the foregoing principle is as follows (see last figure):—

Let P = the initial pressure of steam in lbs. on the square inch at the cylinder, reckoned from absolute zero or perfect vacuum line, or = A C.

p_m = the mean pressure in lbs. on the square inch throughout the stroke, also reckoned from absolute zero.

A = area of cylinder in square inches.

L = whole stroke, A B, in feet.

l = distance in feet to point of cut-off, or C D.

$\frac{L}{l} = r$ = ratio of expansion, neglecting clearance.

x = any distance from commencement of stroke between the limits, $x = l$ and $x = L$.

Then the whole work done through distance, l, = A P l, foot-lbs.

Pressure of steam at any point, x, $= \frac{A P l}{x}$.

$\therefore$ The work done through any very small space $dx = \frac{\text{A P} l}{x} dx$.

The whole work done during expansion—*i.e.*, from point of cut-off to the end of the stroke, or from where $x = l$ to where $x = \text{L}$ is

By integral calculus,

$$= \text{A P} l \int \frac{dx}{x} = \text{A P} l \log._e \frac{\text{L}}{l} = \text{A P} l \log._e r, \text{ foot-lbs.}$$

$\therefore$ The whole work done during one stroke,

$$= \text{A P} l + \text{A P} l \log._e r = \text{A P} l (1 + \log._e r).$$

And the total horse-power, if N = number of strokes per minute,

$$= \frac{\text{A P} l \text{N} (1 + \log._e r)}{33\,000},$$

The total forward mean pressure, p_m, indicated by the vertical height, p_m, is therefore found by dividing the above whole work done during one stroke by the area, A, and by the length of the stroke, L,

$$\text{or } p_m = \frac{\text{A P} l}{\text{A L}}(1 + \log._e r) = \frac{\text{P} l}{\text{L}}(1 + \log._e r) = \frac{\text{P}}{r}(1 + \log._e r).$$

And if p_b = the mean back pressure indicated by the vertical height, p_b, in the last figure, or by the shaded portion above the line, A B; and p, = the net or effective mean pressure throughout the stroke, then—

$$p = p_m - p_b = \frac{\text{P}}{r}(1 + \log._e r) - p_b \text{ lbs. on the square inch.}$$

And the Net or Effective Horse-power

$$= \text{A L N} \frac{\left\{ \frac{\text{P}}{r}(1 + \log._e r) - p_b \right\}}{33000}$$

These formulæ take no account of the wiredrawing of the steam between the boiler and the engine, or in the steam ports, neither have the effects of clearance, compression, &c., been taken into account. They must not therefore be used in determining

the size of any particular engine, because large allowances have sometimes to be made for these effects in actual practice; but as they are sufficient to solve most of the ordinary questions set in examination papers, we shall apply them to three examples in order to impress them on the student's memory, and thus lead up to the final formula.

1st. Take the case of p. 148, Watt's diagram of work. Here P = 1 atmosphere, or say 15 lbs. absolute, for Watt at the time of his devising his diagram of work only used steam of atmospheric pressure, and thus all work was done in his engines at that time, solely by means of the vacuum. The ratio of expansion, $r = 5$, since steam was cut off at $\frac{1}{5}$ of the stroke, and he took no account of back pressure, thus supposing the vacuum to be perfect—

The mean pressure,

$$p_m = \frac{P}{r}(1 + \log._e r) = \frac{15}{5}(1 + 1{\cdot}609). \quad \text{See p. 26? for logs.}$$

$$p_m = 3 \times 2{\cdot}609 = 7{\cdot}827 \text{ lbs., or } {\cdot}52 \text{ of an atmosphere,}$$

which corresponds with that found by Simpson's or ordinary rule (see p. 149).

2nd. Take the case at p. 149, where the pressure of steam may also be supposed to be that above a perfect vacuum and no back pressure was mentioned.

$P = 100$ lbs. absolute, $r = 4$, as steam was cut off at $\frac{1}{4}$ stroke;

∴ mean pressure,

$$p_m = \frac{P}{r}(1 + \log._e r) = \frac{100}{4}(1 + 1{\cdot}386)$$

$$p_m = 25 \times 2{\cdot}386 = 59{\cdot}65 \text{ lbs.,}$$

As against 59·7 lbs. found at p. 150, and 59·9 at p. 151.

3rd. Let us see what we might have expected the mean forward pressure to be in the case of the non-condensing Armstrong engine, whose indicator diagram is shown at p. 229, and calculated horse-power at pp. 261, 262, supposing the boiler pressure to be known, as well as the back pressure, and neglecting clearance. The pressure at the boiler is marked 70 lbs.—*i.e.*, above the atmosphere, or adding the pressure of the atmosphere 15 lbs. we have $P = 70 + 15 = 85$ lbs. The cut-off is at nearly $\frac{1}{3}$ stroke, or $r = 3$, and the back pressure is just 15 lbs., as the exhaust line coincides exactly with the atmospheric line. It is not usual, however, for the exhaust to be so free as this in such engines.

The mean net or effective pressure is—

$$p = \frac{P}{r}(1 + \log_e r) - p_b \qquad \frac{85}{3}(1 + 1\cdot0986) - 15$$

$$= 28\cdot\dot{3} \times 2\cdot0986 - 15 = 59\cdot45 - 15 = 44\cdot45 \text{ lbs.}$$

As against 43·5 lbs. marked on the indicator diagram in Lecture XVI.

We must now take the effect of clearance into account, in order to get a more perfect estimate of the probable mean pressure in any case we may have to deal with in practice.

If the student refers back to Lecture XV., he will see that the ratio of expansion, r, as treated above, becomes r_1 when we take clearance into account, and that

$$r_1 = \frac{r(1 + c)}{1 + c\,r}$$

Where, c, the clearance, is considered as the fraction of the whole volume of the cylinder to the point of cut-off. It will, however, be more convenient here to consider, c, as an addition to the length of the cylinder, the area of this supposed clearance-length, c, being equal to that of the cylinder, = A, so that $c \times A$ = volume of clearance,* and therefore the true ratio of expansion becomes

$$\frac{L + c}{l + c} = \frac{\text{length of stroke} + \text{clearance.}}{\text{length to cut-off} + \text{clearance.}}$$

The clearance is shown in the last figure by the distance, c.

* It is not possible to estimate exactly the volume of the clearance in a completed or working engine, unless the valve casing cover be taken off, the piston brought first to one end of the cylinder, and the volume of water required to just fill the clearance spaces at the end between the piston and right up to the valve face be measured, and then the same operation performed for the other end of cylinder. Of course, it may be calculated approximately from the drawings of the engine, or allowed for in calculations previous to making the drawings. This volume of the combined clearance spaces, at one end or the other, is then considered as a fraction or percentage of the whole volume of the piston's stroke, or it may be regarded as equivalent to a fraction, c, of the stroke, L. For if, A, be the sectional area of the cylinder in square feet, then $A \times L$ = volume of the cylinder's stroke in cubic feet, and $A \times c$ = volume of clearance spaces also in cubic feet.

Hence $A(L + c)$ = whole volume of cylinder, including clearance,
and $A(l + c)$ = whole volume to point of cut-off, including clearance

Therefore, the actual ratio of expansion,

$$= \frac{A(L + c)}{A(l + c)} = \frac{L + c}{l + c} \text{ the expression used above.}$$

Now, reasoning as before—
The *whole* work done to the point of cut-off $= \mathrm{A\,P}l$.
The *whole* work done during expansion

$$= \mathrm{A\,P}\left\{(l + c)\left(\log_e \frac{\mathrm{L} + c}{l + c}\right)\right\}$$

The sum of these two quantities equals the whole work done during one whole stroke, and is

$$= \mathrm{A\,P}\left\{l + (l + c)\left(\log_e \frac{\mathrm{L} + c}{l + c}\right)\right\} \text{ neglecting back pressure.}$$

The mean forward pressure during the stroke is found, by dividing this expression by the area of cylinder, A, and by the length of the stroke, L, and subtracting the mean back pressure, p_b.
Or

$$p_m - p_b = p = \frac{\mathrm{P}}{\mathrm{L}}\left\{l + (l + c)\left(\log_e \frac{\mathrm{L} + c}{l + c}\right)\right\} - p_b$$

Applying this formula to the last example (see also pp. 229 and 262), where P = 85 lbs., being 70 lbs. boiler pressure plus 15 lbs. atmospheric pressure, L = 2 ft., $l = \frac{2}{3}$ ft. (as steam was cut off at $\frac{1}{3}$ stroke), and assuming, c, to be equivalent to $\frac{1}{10}$ of the stroke, or ·2 ft., which is a common allowance, while the back pressure, $p_b = 15$ lbs. (for as we noticed before the exhaust line and the atmospheric line agree), we have by substituting these known values in the last equation—

$$p = \frac{85}{2}\left\{\tfrac{2}{3}' + (\tfrac{2}{3}' + \cdot 2')\left(\log_e \frac{2' + \cdot 2'}{\frac{2}{3} + \cdot 2}\right)\right\} - 15$$

$$p = 42{\cdot}5\,\{\cdot\dot{6} + \cdot 8\dot{6}\,(\log_e\ 2{\cdot}54)\} - 15$$

NOTE.—The nearest log. to 2 54 in the following table is that of 2·5.

$$p = 42{\cdot}5\,(\cdot\dot{6} + \cdot 8\dot{6} \times \cdot 91629) - 15 = 42{\cdot}5 \times 1{\cdot}46 - 15$$

$$p = 62{\cdot}05 - 15 = 47{\cdot}05 \text{ lbs., as against } 44{\cdot}45 \text{ lbs.}$$

by our former formula when not taking clearance into account, and as against 43·5 lbs. on the indicator card. But, as we mentioned before, wire drawing, &c., reduces the pressure between

the boiler and the cylinder, and on looking at the indicator card at p. 229 we observe that the initial pressure on it is marked 65 lbs., or a fall of 5 lbs., or 13·4 per cent., between the boiler and the piston. If we take 65 as the initial pressure, then the total pressure, P, becomes 65 + 15 or 80 lbs., and substituting this value in the last formula for the 85 lbs., we get a mean cylinder pressure of 43·4 lbs., which is certainly a very close approximation to the mean cylinder pressure 43·5 lbs., as found from the actual indicator diagram by measurement. It must be admitted, however, that this indicator diagram is an exceptionally good one, and corresponds more closely in form than most engine diagrams do, to a theoretically perfect diagram.

It is therefore advisable to be cautious in trusting to this formula. It will well repay time spent to draw out to a large scale the most probable indicator diagram for any engine that we may be designing,* bringing to bear any known results for the reduction of boiler pressure due to wire drawing under similar conditions, as well as for the effects of clearance, release, and compression on the area and on the form of the diagram, so as to ascertain the mean pressure, and thereby the horse-power graphically, as well as by the formula; for actual final results as found by indicator diagrams have been known to vary 25 per cent. from the previous calculated results, when trusting merely to the formula and to the supposed boiler pressure. Of course such a result might be fairly termed a miscalculation!

The following Napierian logarithms will facilitate the calculation of mean pressures:—

HYPERBOLIC OR NAPIERIAN LOGARITHMS OF RATIOS OF EXPANSION.

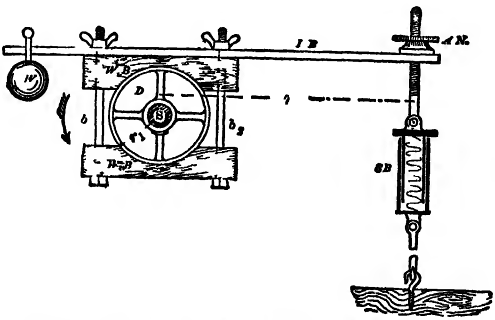
No.	Logarithm	No	Logarithm	No	Logarithm	No	Logarithm.
1	0	3·5	1·2527629	6	1·7917595	8·5	2·1400661
1 25	·2231435	3·75	1 3217559	6·25	1 8325814	8·75	2·1690536
1·5	·4054652	4	1·3862943	6·5	1·8718021	9	2·1972245
1·75	·5596157	4·25	1·4469189	6·75	1·9095425	9·25	2·2246236
2	·6931472	4·5	1·5040773	7	1·9459100	9·5	2·2512918
2·25	·8109303	4·75	1·5581446	7·25	1·9810014	9·75	2·2772673
2·5	·9162907	5	1·6094379	7·5	2·0149030	10	2·3025851
2·75	1·0116009	5·25	1·6582280	7·75	2 0476928	12	2·4849065
3	1·0986124	5·5	1·7047481	8	2·0794414	15	2·7080502
3·25	1·1786549	5·75	1·7491998	8·25	2·1102128	18	2·8903847

* The plan of plotting diagrams to one scale as explained at the end of Lecture XVI., should be followed in the case of compound engines.

Brake Horse-power.*—It is often advisable, more especially in the case of competitive trials of Land and Electric Light Engines, to know the actual power given out by an engine independent of the power absorbed in friction, &c., in driving the engine itself. In order to ascertain this, it is necessary either to apply an absorption or a transmission dynamometer to the fly-wheel, or to a pulley keyed on the crank or first shaft. The power so obtained, is termed the Brake Horse-Power and symbolised by the letters B.H.P.

It is certainly much more satisfactory to the buyer of an engine to know definitely the B H P. of an engine, than either the almost obsolete N.H P., or the now more common I.H.P., for thereby he knows exactly what power he can get from the engine at a certain speed; and it would be well, both for buyers and sellers, if this system of reckoning the power of smaller engines was always insisted upon, and a test made before acceptance.

One of the simplest and most easily applied Absorption Dynamometers is that known as the Prony Brake, which we now illustrate and explain by an actual example of a test made by the author.



PRONY BRAKE OR ABSORPTION DYNAMOMETER.

Where W B	represents	Wooden blocks to fit
D	,,	Drum or pulley keyed to
S	,,	Driving shaft
b_1, b_2	,,	Iron bolts with ram's horn nuts to adjust the tightness of W B on D
I B	,,	Stiff iron bar with
S B	,,	Salter's balance at one end, and
C W	,,	Small counter weight to balance extra length of I B and S B on other side
A N	,,	Adjusting nut for Salter's balance.

* The Student should also refer to the Author's *Text-Book of Applied Mechanics and Mechanical Engineering*, vol. i., for a more complete treatment of this subject.

Method of taking Test for Brake Horse-Power.

1. Adjust position of C W until it balances the weight of I B, A N, and S B, with the wooden blocks slack on pulley.

2. Start machinery and tighten blocks, W B, by ram nuts until desired speed is attained, at same time adjusting S B by nut, A N, until a balance is obtained, keeping I B level.

Note number of revolutions per minute by speed indicator and stress indicated by spring balance.

$$\text{H.P.} = \frac{2\pi r n \text{P}}{33000} \text{ horse-power developed on brake.}$$

Where r = horizontal distance from centre of balance to centre of shaft S in feet.

n = number of revolutions per minute.

P = Salter's balance reading.

$$\text{Since } \frac{2\pi}{33000} = \cdot 0001904 = \text{a constant.}$$

$$\text{H.P.} = \cdot 0001904 \times r \times n \times \text{P}.$$

Ex.—Test recently taken by the author of fast-speed Westinghouse engine (diameter of cylinder 7-inch, stroke 5-inch, pressure of steam 55 lbs.), with crank shaft coupled direct to an Edison dynamo.

The blocks, W B, were fixed to a fly-wheel of 2 feet diameter, which was 6 inches broad.

$$r = 2\cdot 5 \text{ feet}; \; n = 624; \; \text{P} = 48 \text{ lbs.}$$

$$\therefore \text{H.P.} = \cdot 0001904 \times r \times n \times \text{P}$$

$$\therefore \text{H.P.} = \cdot 0001904 \times 2\cdot 5 \times 624 \times 48$$

$$\therefore \text{H.P.} = 14\cdot 26.$$

It is important to note that neither the diameter of the pulley nor the pressure of the friction blocks on the same (due to the weight of the apparatus, or the tightening of the ram nuts), nor the coefficient of friction enter into the formula for obtaining the horse-power. The only data required being the horizontal length of lever, r, the pull, P, and the number of revolutions.

For, let, p, be the pressure, and, f, the coefficient of friction between the face of the drum, D, and two brake blocks, W B, then the twisting moment, T, tending to turn the brake blocks round with the shaft is

$$\text{T} = 2pf \times r_1$$

Where r_1 is the radius of the pulley or drum, D, in feet.

But this twisting moment is balanced by the pull on the spring balance, P, multiplied by its leverage, r.

$$\therefore 2pfr_1 = Pr.$$

The angle turned by the pulley or drum, D, per minute $= 2\pi n$ radians, and since the work done by a couple is the product of its moment into the angle through which the body acted on turns .—

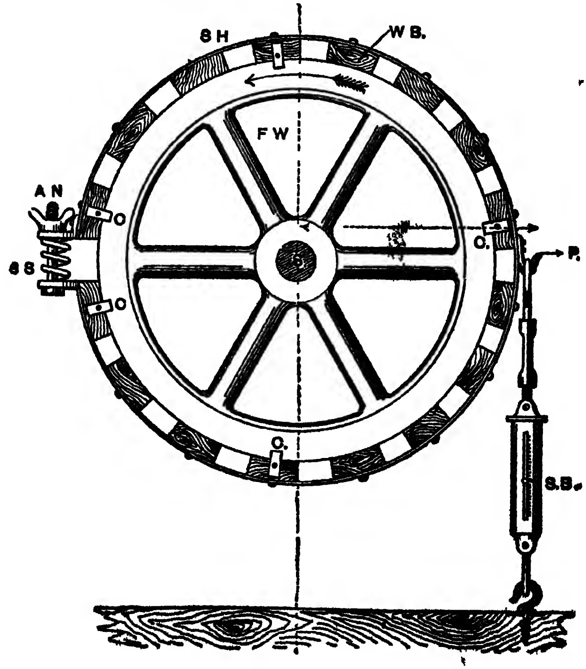
The work absorbed by friction = The work done per minute in foot-pounds, *i e.*,

$$2pfr_1 \times 2\pi n = Pr \times 2\pi n$$

$$\text{and } \therefore \text{ the H P.} = \frac{Pr \times 2\pi n}{33000} = \frac{2\pi r n P}{33000}.$$

It is sometimes advisable to add a dash pot to the lever, I B, in order to get steady readings of the Salter's balance or weight, P.

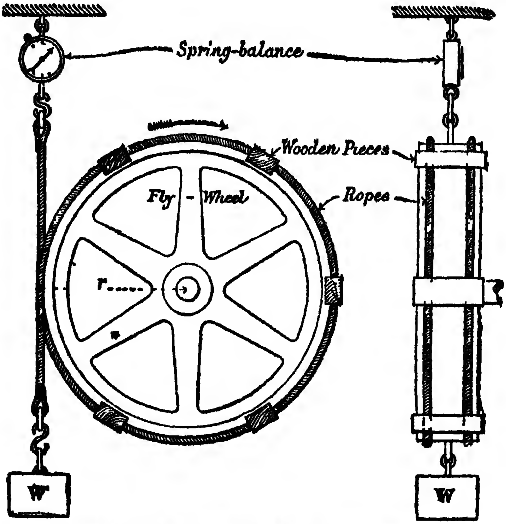
Another very useful and practical form of Prony Brake is that shown in the following figure :—



Here the balance weight and ram nuts are done away with, in favour of a steel hoop or strap, S H, to which are fitted wooden blocks, W B, with spaces of, say, 2 inches or so intervening between them, surrounding the flywheel, F W, keyed on the crank shaft, S. Clips, C, made of iron or steel, keep this brake strap fair on the flywheel, and thus prevent it from sliding to one side more than another.

The engine is started with the adjusting nut, A N, and the spiral spring, S S, slack until it reaches the normal speed. The nut, A N, is now gradually tightened, the speed being kept constant and the pointer, P, level, the tension on the Salter's balance, S B, is read off and the calculation made for the B H P. exactly as in the former example.

Society of Arts Rope Dynamometer.—The jurors for the famous gas engine trials, held under the auspices of the



SOCIETY OF ARTS ROPE DYNAMOMETER.

London "Society of Arts" in 1888, were the first to publicly use a rope-brake in any extensive series of competitive trials, and hence the general name which has been given to this very simple and excellent form of brake. But, rope-brakes had been designed and used prior to these tests by at least four well-known persons. As will be gathered from the

following three sets of figures, this brake consists of an endless flexible rope, doubled round a pulley or the flywheel of an engine, and fitted with several ⊓ shaped wooden distance pieces, in order to keep the two parts of the rope uniformly apart and also to prevent them slipping off the wheel. These distance pieces or clips should be secured to the rope by soft copper wire lacing, drawn in from the outside of the clips and then through the centre of the rope, instead of being fastened thereto by nails or screws from the inside; for such latter metal fastenings are liable to part, to heat, and, consequently, char the rope The rope should be thoroughly stretched and treated with castor oil or grease and black lead powder, prior to its being fitted to the wheel and to the clips, whenever long and important tests are desired. No further lubrication is required, and consequently the first and second defects mentioned on a previous page as pertaining to strap-brakes are entirely avoided. If large powers are to be demanded from a wheel of limited size, then it should have its rim of ⊔ section, so that a small stream of water may be played into the inside of the hollow part of the rim, which water will help very materially by its evaporation to dissipate the heat generated by the friction between the brake rope and the outer surface of the wheel. The surface of the pulley should be flat instead of rounded, in order to get the rope to work perfectly smooth, and a trial run of a few hours prior to the special test is advisable, in order to bring about a small flat glazed surface on the rope, which glazing is materially assisted by the previous application of the black lead powder. For anything up to 5 B.H.P. at 1,000 or more feet per minute of friction surface speed, the author has found that a flexible ship's log-line about ·3 inch in diameter with a double turn round the wheel forms an excellent brake rope. From 5 to 10 B.H.P. a ·5-inch diameter manilla rope serves the purpose. From 10 to 30 B.H.P. a ·6-inch rope will do, and for 100 to 150 B.H.P. (at about 4,000 feet per minute) four turns of 1-inch rope on a large 16 feet diameter flywheel runs quite cool, as may be seen from the next figures on absorption dynamometers in this Lecture.

Advantages of the Rope-Brake.—The author has tested a large number and variety of motors with the rope-brake, and he considers that it has the following advantages:—

1. It can be constructed on short notice, from materials always at hand, in a factory or workshop, and at little expense.

2. It is so self-adjusting that very accurate fitting is not required.

3. It can be put on and taken off the brake-wheel in a very short time.

4. Being comparatively light and of small bulk, it can be hung up on the wall of the testing room, or laid past in a cupboard for future use.

5. It requires no attention whatever for lubrication, if the previously mentioned precautions as to treating and fitting the same are attended to.

6. The back pull registered by the spring-balance may be rendered very steady and of small amount by properly adjusting the weight, W, prior to the commencement of the recorded brake trials.

7. The brake-wheel, if of the proper size, soon attains a maximum temperature, so that the radiated heat equals that generated by the friction.

8. It may be used for very small as well as for large powers.

9. For large powers more and stronger ropes are only required on a comparatively larger wheel, and with the water-cooling device mentioned in the previous section. The greatest power which the author has tested with a rope-brake was an engine of 140 I.H.P. for five continuous hours.

Tests of Small Engines with the Rope-Brake.—Fig. 1 shows the arrangement of dead weight and Salter's balance used by the author in testing gas engines, and Fig. 2 the way in which he applied two spring balances to the brake rope in case of high speed steam engines. The latter plan has, under certain circumstances, particular advantages over the former. By selecting two spring balances with different periods of oscillation, the tendency to jerk or "hunt" may be considerably reduced, or even entirely checked.

RESULTS OF TWO TESTS WITH THE ROPE-BRAKE.

DATA.	"Acme Gas Engine" by Alex Burt & Co., Glasgow.	"Brown's Rotary Engine" by Lang & Sons, Johnstone.
Duration of tests in hours,	4	5
Initial gas or steam pressure in lbs. per sq. in. above atmosphere,	150	95
Final gas or steam pressure in lbs. per sq. in. above atmosphere,	1	1·5
Radius of brake load in feet,	2·771	2·042
Mean revolutions per minute,	154	574·5
Mean nett brake load in lbs.,	231	93·2
Mean B.H.P.,	18·77	20·8
Gas in cb. ft. or steam in lbs. per B.H.P.-hour,	19·13 cb. ft.	37·9 lbs.

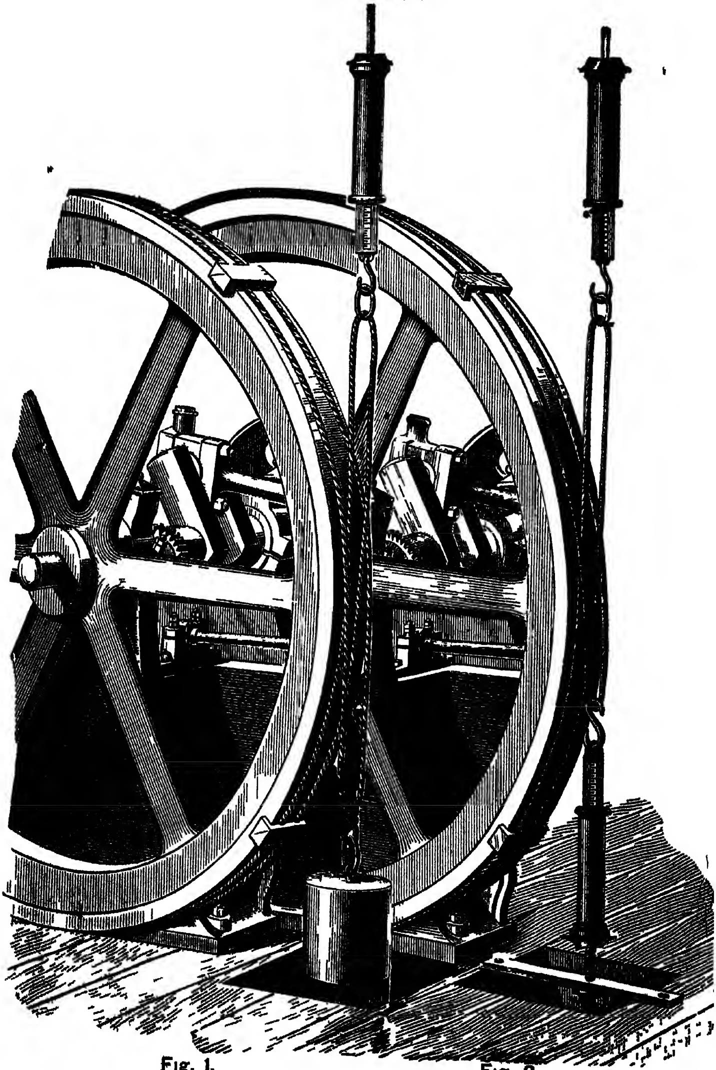


Fig. 1. Fig. 2.

THE TWO FORMS OF BRAKE

USED BY PROF. JAMIESON IN TESTING THE "AJAX" GLASGOW GAS ENGINE FOR BRAKE HORSE-POWER.

MEAN RESULTS OF A THREE HOURS' AND A SIX HOURS' CONTINUOUS B.H.P. TESTS AT FULL NORMAL WORKING POWER; ALSO, HALF-AN-HOUR AT FULL POWER OF THE "AJAX" GLASGOW GAS ENGINE.*

	March 9, 1889, Mean of Three Hours' Tests, 9.20—12.20.	March 29, 1889, Mean of Six Hours Tests, 12 to 6 p.m.						Mean of Previous Six Hours' Tests, 12—6.	March 29, 1889, Full Power Test, 6—6.30 p.m.
		First Hour, 12—1.	Second Hour, 1—2.	Third Hour, 2—3.	Fourth Hour, 3—4.	Fifth Hour, 4—5.	Sixth Hour, 5—6.		
Revolutions per Minute, .	173·5	180·5	180·3	175·3	176·1	175·2	177·3	177·5	200·2
Net Brake Load, in lbs., .	99	98	98	98	98	98	98	98	98
Gas Consumption (*Main only*) in Cubic Feet, per Hour, ...	189·6	184	186	181	181	183	185	183	218
Brake-Horse-Power, . .	8·84	9·1	9·1	8·85	8·9	8·84	8·95	8·96	10·1
Gas, per Brake-Horse-Power, in Cubic Feet, per Hour, ...	21·5	20·2	20·4	20·4	20·3	20·7	20·6	20·4	21·6
Mean Effective Pressure, in lbs., per Square Inch,	60·8	...	...	...	...	...	...	63	...
Indicated Horse Power from above data, . .	10·04	...	...	...	...	...	...	10·6	...
Gas, per I.H.P., in Cubic Feet, per Hour (*Main only*), ...	18·9	...	...	...	...	...	...	17·2	...
Mechanical Efficiency of Engine, or $\frac{\text{B.H.P}}{\text{I.H.P.}} \times 100$,	87·9%	...	...	...	...	...	...	84·2	...

* Students who are interested in this subject of testing prime motors by absorption and transmission dynamometers, will find complete details of a five hours' continuous test of "Field's Combined Steam and Hot-Air Engine," as well as illustrated descriptions of "Froude's Water Dynamometer," in Vol. I. of the author's *Text-Book on Applied Mechanics and Mechanical Engineering*. The latter dynamometer may be used for over 1,000 H.P., and forms a most interesting study of the conversion of work into heat.

Lecture XVII.—Questions.

1. Define the horse-power of an engine. Explain the method adopted for measuring the work actually done in the steam cylinder of an engine. Write down the formula by which the horse-power of an engine is obtained.

2. Having given the indicator diagrams of a steam engine, explain in detail how you would determine the indicated horse-power, and state what additional data are required. (C. & G., 1902, O., Sec. C.)

3. What diameter of cylinder will develop 50 horse-power with a 4-foot stroke, 40 revolutions per minute, and a mean effective steam pressure of 30 lbs. above the atmosphere, the engine being non-condensing? *Ans.* 14·78.

4. The diameter of cylinder of non condensing engine is 18 ins., length of stroke 2 ft. 6 ins., mean pressure of steam 20 lbs. on sq. in. above the atmosphere. Find the number of revolutions per minute when the engine develops 27 H.P. *Ans.* = 35 revs. per min.

5. Having obtained indicator diagrams from a single cylinder engine, state particularly, and with the necessary details and sketches, the method of obtaining the indicated horse-power. A motion of 1 inch in the pencil of the indicator, as due to steam pressure = 20 lbs. on the sq. inch, what is the H.P. at 90 revolutions per minute, if diameter of the piston be 10 inches, stroke 20 inches, area of diagram 8 sq. inches and length 5 inches? *Ans.* 22·8.

6. The two cylinders of a locomotive engine are each 17 inches in diameter, and the length of stroke is 24 inches, also the driving wheel makes 100 revolutions per minute, and the mean effective pressure of the steam is 80 lbs. Find the horse-power. *Ans.* 440·3.

7. The diameter of the cylinder of an engine being 53 inches, the stroke 5 feet, and the number of revolutions 30 per minute, find the mean pressure of the steam to develop 600 indicated horse power. *Ans.* 29·9.

8. In a compound cylinder marine engine, the diameter of the high-pressure cylinder is 57 inches, and that of the low-pressure cylinder is 100 inches, the stroke of each piston being $2\frac{3}{4}$ feet. The mean pressures of the steam in the respective cylinders are 26 lbs. and $8\frac{1}{2}$ lbs., and I.H.P. is 1,028; find the number of revolutions made in one minute. With what view is an engine constructed in the manner pointed out? *Ans.* 46·3.

9. In a compound cylinder tandem engine the steam is cut off at $\frac{1}{3}$ of the stroke in the high-pressure cylinder, the areas of the pistons are as 1 to 3, and the diameter of the smaller cylinder is 20 inches; investigate an expression for the work done in one stroke. Example: Find the horse-power of the engine when the initial pressure of the steam is 85 lbs. per square inch above that of the atmosphere—viz., 15 lbs., the back pressure in the large cylinder is 3 lbs. per square inch, and the speed of each piston is 300 feet per minute. *Ans.* 278 taken isothermally.

10. An engine gives 10 indicated horse-power and 7·6 brake horse-power for a consumption of 230 cubic feet of gas per hour. Explain how these measurements are made. If the calorific power of 1 cubic foot of the gas is 530,000 foot-pounds, what is the efficiency? What is your notion of how all the energy of 1 cubic foot of gas is disposed of? (S. & A., 1897, Adv.)

11. Initial pressure of steam 180 lbs. per square inch, back pressure 17 lbs. per square inch, cut off at one-third of the stroke, area of piston 112 square inches, and length of crank 1 foot; what work is done in one stroke? What is the weight of steam used in one stroke if the volume of 1 lb. of the steam is 2·51 cubic feet? If there are 200 strokes per minute, what is the indicated horse-power and what weight of steam is used per hour, neglecting clearance, condensation, and leakage? (B. of E., 1900, Adv.)

12. Steam of 150 lbs. per square inch (absolute) is cut off at one-third the stroke, and expands according to the law pv constant. Find the average pressure in the forward stroke, using squared paper. The back pressure is 18 lbs. per square inch, what is the effective pressure on the piston? The piston is 15 inches diameter, crank 1 foot, two strokes in the revolution, 120 revolutions per minute; find the work done in one revolution and also the horse-power. One lb. of steam of 150 lbs. pressure has a volume of 2·978 cubic feet, what weight of steam is indicated per hour? (B. of E., 1902, Adv.)

13. Steam is cut off at one-third the stroke and expands by the law pv constant; find the average pressure in the forward stroke as a fraction of p_1, the initial pressure. The back pressure is 18 lbs. per square inch, together with 10 lbs. per square inch representing the friction of the engine. The piston is 15 inches diameter, crank 1 foot, two strokes per revolution, 120 revolutions per minute; find the actual horse-power for each of the three values of p_1 given in the table below. One lb. of steam pressure, p_1, has the volume, v_1 cubic feet, given in the table; what weight of steam, W, is indicated per hour in each case? Show, on squared paper, the probable relationship of W and the horse-power in this engine when p_1 varies but the cut-off of one-third is not altered.

p_1	150	120	80
v_1	2·987	3·671	5·37

(B. of E., 1902, H., Part i.)

14. The expected mean steam pressure for an engine, which has to be designed, is 41 lbs. per square inch, and the piston speed adopted is 600 feet a minute: what diameter of cylinder must be adopted to secure an effective H.P. of 78? Assume the effective power is 89 per cent. of that developed in the cylinder. (C. & G., 1902, O., Sec. C.)

15. A steam engine has a cylinder 13 inches in diameter, and a piston speed of 650 feet a minute, calculate its probable indicated H.P. from the following data as to the steam:—Boiler pressure 125 lbs. absolute, cut off at 18 per cent. of piston stroke, and back pressure 16 lbs. absolute. You may neglect compression and release effects, and assume that the clearance volume at each end is $2\frac{1}{2}$ per cent. of the volume swept by the piston. Given $\log_e 5 = 1·609$. (C. & G., 1900, H., Sec. B.)

16.* The area of the piston of an engine is 3 square feet, the pressure of the steam is 15 lbs. per square inch above the atmosphere on admission, and the steam is cut off at $\frac{1}{3}$ of the stroke; the crank shaft makes 40 revolutions per minute, and the length of the stroke is 3 feet, find the H.P. (given hyp. log. 3 = 1·0986124). *Ans.* Exhausting at zero pressure = 65·9.

17.* The cylinder of an engine is 3 feet 6 inches in diameter, the stroke is 5 feet, and the steam is cut off at $\frac{1}{3}$ of the stroke. If steam be admitted into the cylinder at 45 lbs. pressure, find the work done in one stroke (log. 3 = 1·0986124). *Ans.* 218,061 ft.-lbs.

18. Steam enters a cylinder at 80 lbs. absolute, and is cut off at $\frac{1}{3}$ of the stroke. The diameter of the piston is 40 inches, and the length of stroke 5 feet, the number of revolutions being 50 per minute. Back pressure 3 lbs. absolute, find the horse-power of the engine. *Ans.* 1,009.

* See page 280 for solutions.

19. The stroke of a piston is 5 feet, and its diameter is 4 feet, steam is admitted at 20 lbs. absolute (no back pressure), and is cut off at $\frac{1}{3}$ stroke, find work done in one stroke. If steam be cut off at $\frac{1}{4}$ stroke, and the final pressure is required to remain unchanged, what should be the diameter of the cylinder in order that the work done may also remain unchanged? (log 2 = ·6931472, log 3 = 1·0986124). *Ans.* 153,192 ft.-lbs.; 43·1 ins.

20. Explain by a sketch and index, a rope-brake dynamometer. State how it is used and enumerate its advantages.

Having frequently found that students experience a difficulty in working out such questions as Nos. 16 and 17, I have thought it advisable to give their solution in full so that they may the more readily understand how to do similar questions.

Question 16 of Lecture XVII

Given, A = Area of piston = 3 sq. ft. = 3 × 144 = 432 sq. ins.

P = abs. press. per sq. in. = 15 + 15 = 30 lbs.

$$l = \text{point of cut off} = \frac{\text{stroke of engine}}{\text{ratio of expansion}} = \frac{L}{r} = \frac{3\text{ ft.}}{3} = 1\text{ ft.}$$

N = No. strokes per minute = (revolutions) × 2 = 40 × 2 = 80.

$\log_e r = 1{\cdot}0986124.$

$$\text{By formula, H.P.} = \frac{A\,L\,N\left\{\frac{P}{r}(1+\log_e r) - p_b\right\}}{33000}$$

Substituting numerical values—

$$\text{H.P.} = \frac{432 \times 3' \times 80\left\{\frac{30\text{ lbs.}}{3}(1 + 1{\cdot}0986) - 0\right\}}{33000}$$

$$\therefore\ \text{H.P.} = \frac{432 \times 3 \times 80 \times 10 \times 2{\cdot}0986}{33000} = 65{\cdot}9.$$

N.B.—No value being given, p_b, it is assumed as = 0 lbs.

Question 17, Lecture XVII.

Given, d = diameter of cylinder = $3\frac{1}{2}$ ft. = 42 ins.

$$\therefore\ A = \pi r^2 = \frac{22}{7} \times 21 \times 21 = 1386 \text{ sq. ins.}$$

P = 45 lbs. (assumed as the total pressure).

L = length of stroke = 5 ft.

r = ratio of expansion = 3.

Required—The work done in one stroke.

By formula—

$$\text{Mean pressure} = \frac{P}{r}(1+\log_e r) - p_b \text{ in lbs. per sq. inch.}$$

No value being given for p_b, it is assumed as = 0.

Work done = space passed through × force applied.

$$\text{,,} \quad = L \times \frac{P}{r}(1+\log_e r) \times A.$$

Substituting numerical values—

$$\text{Work done} = 5 \times \frac{45}{3} \times (1 + 1{\cdot}0986) \times 1{,}386$$

$$\text{,,} \quad = 5 \times 15 \times 2{\cdot}0986 \times 1{,}386$$

$$\text{,,} \quad = 218{,}150 \text{ foot-lbs. nearly.}$$

21. Describe a method of obtaining the brake horse-power of an engine, and state the advantages to buyer and seller of adopting this method over that of nominal or indicated horse-power. An engine is making 150 revolutions per minute, the diameter of the brake pulley being 4 feet, and the pull on the brake 50 lbs., what is the B.H.P.? *Ans.* 2·85.

22. The diameter of a steam cylinder is 8 inches, the stroke of the piston is 18 inches, the number of revolutions per minute is 150, and the mean effective pressure of the steam is 35 lbs., find the I.H.P., taking $\pi = 3\frac{1}{7}$. The same engine is tested by a brake-pulley on the crank-shaft 5 feet in diameter, the effective load on the brake being 294 lbs., with a radius of $2\frac{1}{2}$ feet. Find the brake horse-power, and the working efficiency of the engine. *Ans.* 24; 21; 87·5 per cent.

23. How would you set about testing an engine? [Choose either a gas or an oil engine, or a steam engine governed by throttling.] Measuring the steam or gas or oil used and the indicated and brake horse-power, what sort of results would you expect to obtain for three tests under steady load of one of these engines? It will be well to give a rough notion of the values actually obtained in any set of tests which you have seen, or of which you have read. (S. & A., 1898, H., Part i.)

24. Steam at 120 lbs. per square inch (absolute), cut-off at one-fourth of the stroke; expansion curve, $p\,v$ constant; back pressure 3 lbs. per square inch. Find the average pressure during the stroke, by calculating the pressures at a number of places. No cushioning, no clearance, release exactly at end of stroke. If the piston is 18 inches diameter, the crank 1 foot, speed 100 revolutions per minute, and the engine double-acting, what is the horse-power? What volume of steam is admitted at each stroke? Also, how much per hour? If this steam measures 3·671 cubic feet to the lb., what is the weight of steam required per hour? If the steam condensed on admission is 40 per cent. of all that is supplied, what is the weight of steam required per hour? (S. & A., 1898, Adv.)

25. With steam at 120 lbs. per square inch (absolute), cut-off at one-fourth of the stroke; expansion curve, $p\,v$ constant. Back pressure 3 lbs. per square inch. No cushioning, no clearance, release exactly at the end of the stroke. The piston is 18 inches diameter, crank 1 foot, speed 100 revolutions per minute, and engine double-acting. If the Napierian logarithm of 4 is 1·3863, what is the hypothetical horse-power? What volume of steam is admitted at each stroke? Also, how much per hour? This steam measures 3·671 cubic feet to the lb. What is the weight of steam per hour? If the steam condensed on admission is 40 per cent. of all that is supplied, what is the weight of steam per hour? Repeat this calculation for steam whose initial pressure is 40 lbs. per square inch, the volume of 1 lb. being 10·4 cubic feet. Plot water per hour, and indicated horse-power, and assume that a straight line connects your points. From this, what is the water per hour when there is no indicated horse-power? (S. & A., 1898, H., Part i.)

26. Sketch and describe any good form of steam engine indicator, and explain what data you require in order to calculate the H.P. of an engine, besides the indicator card. (C. & G., 1900, O., Sec. C.)

27. Take a hypothetical indicator diagram—no clearance, constant back pressure 17 lbs. per square inch. Let friction of engine be represented by 10 lbs. per square inch on the piston. Expansion law $p\,v$ constant. Cut-off at one-third of the stroke. Area of piston 100 square inches, crank 1 foot, 200 working strokes per minute. Steam of the following initial pressures being admitted, find in each case the crank-shaft horse-power, and the

weight of indicated steam per hour. Tabulate the results, and plot upon squared paper. The following information is given:—

Absolute pressure of admitted steam, lbs. per sq. in.,	50	100	150
Cubic feet of 1 lb. of admitted steam, . . .	8·24	4·356	2·978

(B. of E., 1901, H., Part i.)

28. A steam electric generator on three long trials, each with a different point of cut-off on steady load, is found to use the following amounts of steam per hour for the following amounts of power:—

Lbs. of steam per hour, . . .	4,020	6,650	10,800
Indicated horse-power, . . .	210	480	706
Kilowatts produced, . . .	114	290	435

Find the indicated horse-power and the weight of steam used per hour when 330 kilowatts are being produced. Find in the four cases the amounts of steam used per Board of Trade unit (that is, per kilowatt hour). In what way does regulation by varying cut-off differ as to economy of steam under varying load factors, from regulation by varying the pressure letting the cut-off remain constant? (B. of E., 1901, H., Part i.)

29. A test of a small steam motor and its boiler (in which the fuel used was petroleum) gave the following results:—(1) Weight of petroleum used per hour in boiler, 5·53 lbs.; (2) brake H.P. of motor, 3·75; (3) heating value of the petroleum per lb., 20,400 British thermal units; what is the thermal efficiency of the whole plant? (C. & G., 1901, O., Sec. C.)

30. The two cylinders of a locomotive are together found to give an indicated H.P. of 432, and to require 7,560 lbs. of steam per hour. The feed temperature is 60° F. and the boiler temperature 361° F. Assuming that the efficiency of the boiler is 71 per cent., how many pounds of coal per hour are being burnt in the furnace? (The total heat in a pound of steam above 32° F. = 1,082 + 0·3 t thermal units, where t° is the temperature of the steam; and the total heating value of 1 lb. of the coal used = 12,750 British thermal units.) What is the thermal efficiency of the cylinders? (C. & G., 1901, O., Sec. C.)

31. Use the common hypothetic indicator diagram; expansion curve "pv constant"; no clearance or cushioning. A piston is 1 square foot in area, stroke 2 feet, 200 strokes per minute; find the indicated horse-power if the initial pressure of the steam is 120 lbs. per square inch. Take two cases—one in which the cut-off is at half stroke, the other in which the cut-off is at one-fifth of the stroke. This steam is initially 3·67 cubic feet per lb.; find in each case the weight of steam used per hour. It has been found by observation that in the factory driven by the engine the number of yards of stuff made per hour is 7·8 I – 320, where I is the indicated horse-power. Find the number of yards for each of your two cases. Tabulate your answers. State also the number of yards per lb. of steam in each of the cases. (B. of E., 1903, H., Part i.)

32. In testing a double-acting steam engine by a rope brake dynamometer, the diameter of the flywheel was 10 feet, the revolutions 100 per minute, the dead load on the break 1,000 lbs., and the mean pull in the spring 100 lbs. The stroke of the engine was 30 inches, the diameter of the piston 15 inches, and the mean effective pressure 41·2 lbs. per square inch. Find the indicated and the brake horse-power, and also the mechanical efficiency of the engine. (C. & G., 1903, O., Sec. C.)

33. Show, by means of a diagram, how the proportion of the indicated horse-power developed in the respective cylinders of a compound engine may be varied by altering the point of cut-off in the low-pressure cylinder. The ratio of the piston areas of a compound engine is 4 to 1, the initial pressure in the high-pressure cylinder being 105 lbs. per square inch absolute, and the back pressure in the low-pressure cylinder 4 lbs. absolute. Steam is cut off in both the high- and low-pressure cylinders at ·6 of the stroke. Find the ratio of the powers developed and also of the initial forces on the pistons in the two cylinders. Clearance may be neglected, the expansion curve assumed hyperbolic, and the admission and back pressures constant throughout. (C. & G., 1903, H., Sec. B.)

LECTURE XVIII.

CONTENTS.—Action of the Crank—Tangential and Radial Forces—Diagrams of Twisting Moments with Uniform and with Variable Steam Pressure on Piston, neglecting as well as taking Account of the Obliquity of Connecting-rod—Effect of Inertia of Moving Parts—Case of a Horizontal Engine with Connecting-rod of Infinite Length—Example I.—Indicator Diagrams as modified by Inertia—Graphic Representation of the Inertia—Case of a Horizontal Engine with Connecting-rod of Finite Length—Example II.—Position of Instantaneous Axis of Connecting rod—Crank Effort Diagrams of "The Thomas Russell Engine" and of a "Triple-Expansion Engine"—Crank Effort Diagrams of the Quadruple Expansion Five Crank Engines of S.S. "Inchdune"—Example III.—Questions.

Action of the Crank—Tangential and Radial Forces.—In most steam engines the conversion of the reciprocating motion of the piston into circular motion is effected by means of the crank and connecting-rod.

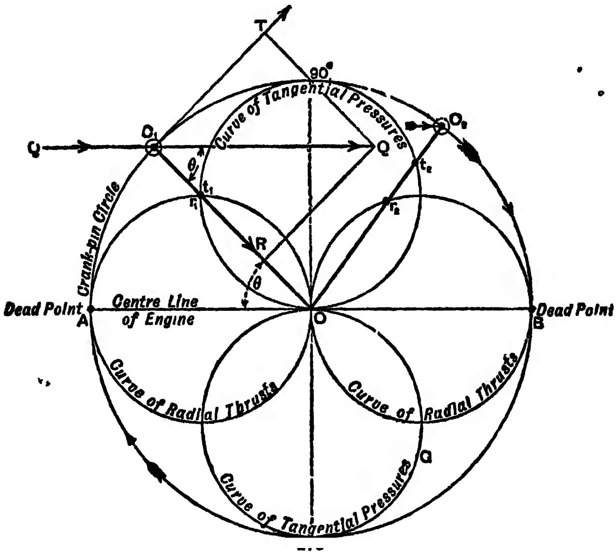
The turning or tangential force exerted by the connecting-rod on the crank varies with the position of the crank itself. Thus, when the centre line of the crank coincides with a line drawn through the centre of the cylinder and the centre of the crank shaft, the crank is said to be at the "dead points," and the connecting-rod exerts no rotational effort on it. The crank arrives in those positions twice in one revolution, just when on the point of reversing the direction of motion of the piston. These positions are O A and O B in the next diagram. Again, when the crank is at an angle of about 90° to the centre line through the cylinder and crank shaft, the tangential force is a maximum.

Diagram of Twisting Moments—Neglecting Length of Connecting-Rod.—The simplest case is that in which the pressure on the piston is uniform throughout the stroke, and the obliquity of the connecting-rod is neglected. Then the pressure or thrust, Q, on the connecting-rod is equal to the total pressure, P, on the piston (see next figure).

Suppose the crank to be in the position, $O C_1$, then by the parallelogram of forces the thrust, Q, on the connecting-rod may be resolved into two components, one, $C_1 R$, acting along the line of the crank and representing a radial pressure, R, on the crank-shaft bearing; the other, $C_1 T$, acting at right angles to $O C_1$, and representing the tangential pressure, T, acting on the crank pin. Of course, the *whole* thrust on crank-shaft bearing is equal to the *whole* pressure on piston.

Let the angle $A O C_1 = \theta, \therefore \angle Q C_1 O = \theta, \because C_1 Q$ is $\parallel$ to $A O$.
Then the radial pressure, R, or $C_1 R = Q . \cos \theta$.
And the tangential pressure, T, or $C_1 T = Q . \sin \theta$.

These components may be plotted out separately for every position of the crank by curves in the following manner:—Let OC_1 represent Q, to any convenient scale, and lay off to the same scale $O t_1 = T = C_1 T$, the tangential component of Q. Then t_1 is a point on the curve, and $O t_1$ measures to scale the tangential pressure on the crank pin for the position, OC_1, of the crank. To find other points on this curve, take any other position of the crank and plot off along that line of the crank the tangential component of Q for that position. If we find a number of points and join them, they



POLAR CURVES OF TANGENTIAL FORCE (T) ON CRANK PIN, AND RADIAL THRUST (R) THROUGH CRANK, WITH UNIFORM PRESSURE ON PISTON AND NEGLECTING OBLIQUITY OF CONNECTING-ROD.

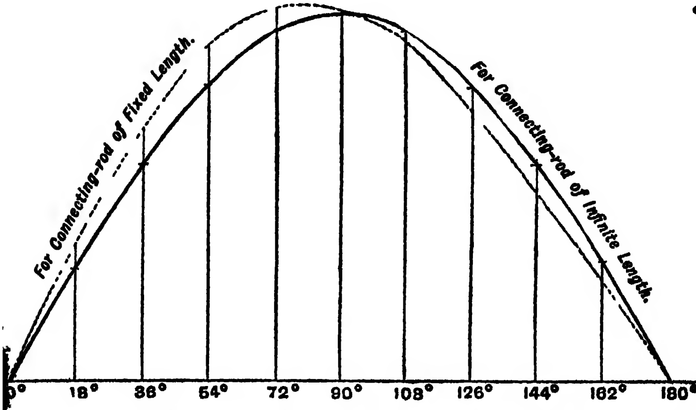
will be found to lie on the circumference of two circles described with O to 90° and O to 270° as diameters. Similarly, if we lay off $O r_1$ on the position, OC_1, of the crank, equal to the radial component of Q, for that position of the crank, and do the same for a number of other positions, we have, by joining the points, two complete circles described with OA and OB as diameters, representing the radial thrust on the crank-shaft bearing for any position of the crank. In the position of the crank taken ($\theta = 45°$) the radial and tangential components are equal, and, therefore, r_1 coincides with t_1. These circles are known as "*Polar*" curves For any other position, OC_2, of the crank, the tangential or turning force is given by $O t_2$, whilst the radial thrust on the crank shaft is given by $O r_2$.

The TWISTING OR TURNING MOMENT OR TORQUE on the crank shaft at any *position is equal to the tangential pressure on the crank pin in that position multiplied by the length of the crank.*

Let r = radius of crank-pin circle or the length of the crank.
,, θ = the angle made by the crank with the line of dead points.
Then the twisting moment = $Q r \sin \theta$.
Or, ,, ,, = $P r \sin \theta$ (for in this case $P = Q$).

Since the polar curves, O to 90° and O to 270°, represent the tangential forces ($P \cdot \sin \theta$), their values must be multiplied by r, the length of the crank, in order to find the TWISTING MOMENT at any point; but, seeing that, r, is constant, the polar curves may be taken to represent the *relative* values of the twisting moments.

The twisting moments may also be represented by the following diagram, in which the horizontal line represents the path of the crank, and the height of each vertical ordinate gives the tangential force or the twisting moment for that point. To draw the diagram for one stroke of the piston, or one half revolution of the crank, lay off a horizontal line equal to the semicircumference of the crank-pin circle, and divide it into 10 equal parts. Each division then represents a movement of $180 \div 10 = 18°$ of the crank.



DIAGRAMS OF TWISTING MOMENTS FOR ONE HALF REVOLUTION OF CRANK.
Both Curves are Drawn on the Assumption of Uniform Pressure on Piston.

Then calculate by the above formula, or plot out by the previous diagram of polar curves, tangential pressures for each of the 10 positions of the crank, and lay them off vertically at each division. Join these points, and we have the above full line curve which represents the twisting moments for one half revolution, neglecting the obliquity of the connecting-rod, and when the pressure on the piston is uniform throughout. A curve for the radial thrust through crank could be plotted out in the same way.

In the early days of the steam engine, it was imagined that the use of the crank and connecting-rod involved a considerable loss of the work developed on the piston. The fallacy of this idea may now be made clear.

The pressure on the crank-pin in the direction of rotation is = $P \sin \theta$;

therefore, in order to obtain the mean tangential pressure during a half revolution of the crank, we have only to find the mean value of sin θ, and multiply it by, P, the total mean pressure on the piston.

For an approximate result take the value of sin θ at every 10 degrees of the crank's movement and divide the total by 18, the number of divisions taken, thus—

Sin	10°	·173
,,	20°	·342
,,	30°	·500
,,	40°	·643
,,	50°	·766
,,	60°	·866
,,	70°	·939
,,	80°	·985
,,	90°	1·000
,,	100°	·985
,,	110°	·939
,,	120°	·866
,,	130°	·766
,,	140°	·643
,,	150°	·500
,,	160°	·324
,,	170°	·173
,,	180°	000
Total		11·428

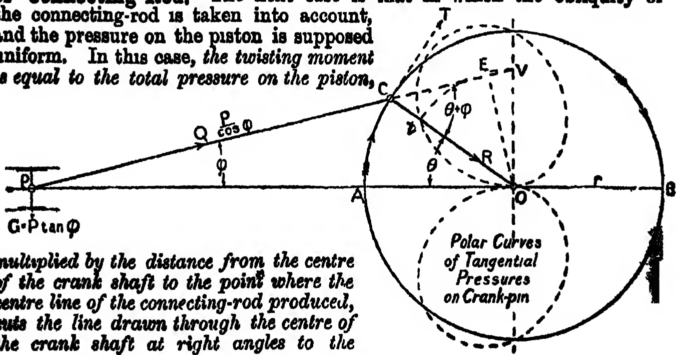
$$\therefore P \times \frac{11{\cdot}428}{18} = P \times {\cdot}6349 = \text{(mean pressure).}$$

Hence, if L = length of stroke = $2r$.
The work done on the crank in one revolution
= Total mean pressure × distance passed through,
$= P \times {\cdot}6349 \times 2\pi r,$
$= P \times {\cdot}6349 \times 3{\cdot}1416 \times L = 1{\cdot}9946\ PL.$

Which is practically the same thing as 2 P L. But, the work done on the piston during one revolution is also equal to 2 P L. Consequently the employment of the crank and connecting rod involves no loss of power if we neglect the power absorbed by friction due to bearing surfaces, &c.

NOTE —No such combination of mechanism as the crank and connecting-rod can involve a loss of power (neglecting friction), as it would be contrary to the "principle of the conservation of energy"

Diagram of Twisting Moments—Taking Account of Length of Connecting-Rod.—The next case is that in which the obliquity of the connecting-rod is taken into account, and the pressure on the piston is supposed uniform. In this case, *the twisting moment is equal to the total pressure on the piston,*



multiplied by the distance from the centre of the crank shaft to the point where the centre line of the connecting-rod produced, cuts the line drawn through the centre of the crank shaft at right angles to the piston's motion.

DIAGRAM OF TWISTING MOMENTS—TAKING ACCOUNT OF LENGTH OF CONNECTING-ROD.

NOTE TO FIGURE —The pressure along connecting-rod and on the crosshead guides may be found graphically for any position, thus—
Let PO = P, the pressure on piston to any convenient scale.
Then PV = Q, the direction and pressure along connecting-rod to the same scale.
And OV = G, the direction and pressure on the lower crosshead guide to same scale.

But $PV \cos\phi = PO$
Or $Q \cos\phi = P$
$\therefore Q = \frac{P}{\cos\phi}$

And $\frac{OV}{OP} = \frac{G}{P} = \tan\phi$
$\therefore G = P\tan\phi$

To prove this, let O be the centre of the crank shaft, and O P the centre line of the engine, passing through, O, and the centre of the cylinder. Let O C be the position of the crank, and P C the length of the connecting-rod. Produce P C to cut the vertical through O in the point V, and draw O E perpendicular to P V. Then $\angle$ V E O = $\angle$ P O V; also $\angle$ P V O is common:

$\therefore$ $\angle$ V O E = $\angle$ O P V = ϕ, the inclination of the connecting-rod to centre line of engine.

$$\text{Now the twisting moment} = Q \times OE = \frac{P}{\cos\phi} \times OV\cos\phi = P \times OV.$$

Knowing this, we can readily construct the polar curves. Suppose the crank in the position, O C, produce the centre line of the connecting-rod to cut the line O V in V. With centre, O, and radius, O V, describe the arc, V t, cutting O C in t. Then, t, is a point on the tangential pressures or twisting moment's curve, and the twisting moment for the position, O C, of the crank is thus P $\times$ O t. A similar construction for all the other positions of the crank gives all the other points, and the complete curve may then be described by joining them. We see that the curve is not a circle as in the last case, but differs therefrom in a marked degree. Now plot off the twisting moment at each of the 10 different points by this method on the rectangular diagram (page 287) as before, and we get the dotted line which shows the new diagram of twisting moments. It will be noticed that this curve rises more abruptly during the first quarter of a revolution, and falls flatter during the second quarter than when the obliquity of the connecting-rod is neglected, thus indicating a greater pressure on the crank pin during the first half of the stroke; also, the maximum pressure is reached before half stroke.

We can calculate the several twisting moments in this case without the aid of a scale diagram, thus—

$$\text{The pressure } Q = \frac{P}{\cos\phi}, \text{ also the angle } OCV = \theta + \phi,$$

Since OCP + OCV = 2 right angles, and OCP + θ + ϕ = 2 right angles

$$\therefore \left.\begin{array}{r}\text{tangential pressure on crank}\\ \text{pin} = T = Q \cdot \sin(\theta+\phi)\end{array}\right\} = \left(\frac{P}{\cos\phi}\right)\sin(\theta+\phi) = P\frac{\sin(\theta+\phi)}{\cos\phi},$$

$$\therefore \text{ the twisting moment} = Pr\frac{\sin(\theta+\phi)}{\cos\phi}.$$

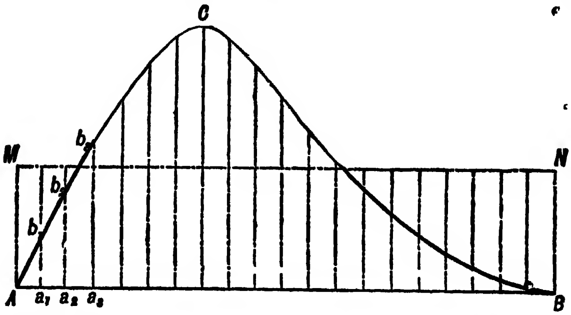
Where $r \sin\theta = l \sin\phi$, r being crank radius, and l the length of connecting-rod.

It is, however, more tedious to work out the results by this formula than by the previous graphic method.

The effect of shortening the connecting-rod is to increase the effort upon the crank pin at the beginning of the stroke, and to decrease it towards the end, thus causing greater irregularity in the tangential pressure on the crank, and greater stress on the crosshead guides. The pressure on the latter is = G = P $\cdot$ tan ϕ, as seen from the last figure and the footnote below it.

The actual state of things which takes place in practice is, however, not so easily represented, for the pressure on the piston is never uniform, but falls away from the point of cut-off. In order, therefore, to construct a true diagram of the twisting moments, we must find the positions of the

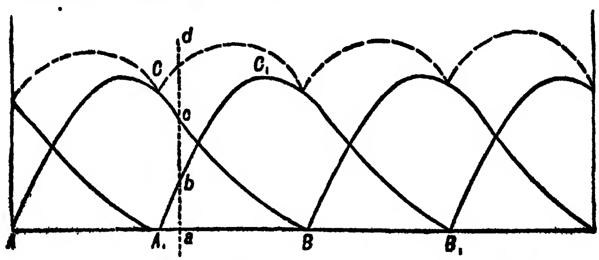
piston corresponding to the various positions of the crank by diagram (Lecture XIV.), and mark these off on the indicator diagram. The steam pressure for the several positions of the crank can then be read off, and their values inserted for, P, in the equation P × O V or in $P\, r \sin(\theta + \phi) \div \cos\phi$. The curve of twisting moments on the crank due *merely* to steam pressure on the piston may then be constructed, as shown by the following diagram:—



CURVE OF TWISTING MOMENTS, TAKING ACCOUNT OF THE VARIATION OF THE STEAM PRESSURE IN THE CYLINDER, AND WITH A CONNECTING-ROD OF KNOWN LENGTH.

On comparing this curve with the two previous curves, it will be seen that between the points, C and B, it falls much lower; this is due to the fall of steam pressure during expansion The rectangle, A M N B, is of the same area as the figure, A C B, and, therefore, A M represents the *mean* twisting moment due to steam pressure on the piston.

When the engine has two cylinders having their pistons working on separate cranks, the curve of total twisting moments on the crank shaft can only be obtained by combining the curves of twisting moments for each crank. This is shown in the following figure, which represents the combined twisting moments on the crank shaft of an engine with two cranks at right angles to each other:—



CURVE OF COMBINED TWISTING MOMENTS FOR TWO CRANKS AT RIGHT ANGLES TO EACH OTHER.

$A C B$ is the curve of twisting moments on one crank, and $A_1 C_1 B_1$ the curve of twisting moments on the other crank during one-half revolution, the remaining curves being for the other half revolution. To find the total twisting moment at any point, a, draw the vertical line $a d$, and make $a d = a c + a b$ (*i.e.*, the sum of the twisting moments on each crank). By finding a number of points in this way, the whole curve of total twisting moments may be plotted out.

Effect of Inertia of Moving Parts.—In finding the twisting moments by these methods, we have neglected a most important effect—viz., the variation of effort on the crank shaft due to the inertia of the moving parts. Since the piston is brought to rest at the end of each stroke, the inertia of the piston, piston-rod, crosshead, and connecting-rod, has to be overcome at the beginning of each stroke, in order to start the motion, and a portion of the energy of the steam is absorbed in doing this; therefore, the actual effort on the crank in the first half of the stroke is *less* than that given by the curves. The energy which is imparted to the moving parts is, however, given out on the crank during the latter part of the stroke, when these moving parts are being brought to rest; therefore, the effort on the crank during the second half of the stroke is *greater* than that shown by the curves. On the principle of the conservation of energy this alternate acceleration and retardation can neither add to, nor subtract from, the total power developed during the stroke. In ordinary cases, therefore, the inertia of the moving parts acts as a fly-wheel would do, and tends to equalise the effort on the crank. The effect, however, at different parts of the stroke is very interesting and instructive, especially when high-initial pressure and a large range of expansion are adopted, combined with heavy and quickly-moving parts. Allowance may be made for this inertia, if the weight of the moving parts and their velocity are known. We can make an alteration on the indicator diagram, reducing the effective pressure at the beginning of the stroke and increasing it at the end. The steam and the inertia stresses can, however, be combined, only so far as some of the effects are concerned. They are combined, of course, in their pressure on the crank pin, &c., but since the dynamical stresses are not taken up altogether by the engine framing, provision must be made for transmitting them to the engine foundation. These dynamical stresses, introduced by arresting the momentum of the moving parts, produce a much more serious effect in fast-running engines than is usually supposed.

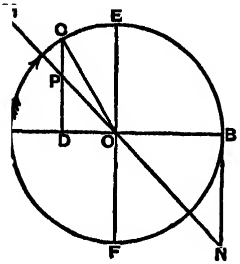
Effect of Inertia on Moving Parts of a Horizontal Engine with Connecting-Rod of Infinite Length.

The student should now try to follow the following investigations and practical examples.

All reference to trigonometry, co-ordinate geometry, and the differential calculus has been dispensed with; and any student with a knowledge of elementary geometry and mechanics will, by carefully following the argument, grasp this subject sufficiently for all practical purposes.

It may be well to state that we shall not take into account *all* the effects resulting from the conversion of reciprocating into circular motion by means of a crank and connecting-rod, but only those effects which directly influence the propelling force on the piston of the engine,* and unless otherwise stated, we shall suppose the engine to be horizontal.

The first case is that in which the motion of the piston is the same as if the engine worked with a connecting-rod of infinite length, such as, for example, in the common donkey engine arrangement where the crank-pin works in a slotted cross-head at right angles to the direction of the piston's motion.



In the figure, let O, be the centre of the crank shaft, O C, the crank, and, A B, the centre line of a horizontal engine of stroke, A B.

From any position of the crank-pin, C, draw the ordinate, C D, at right angles to A B.

Then when the crank-pin is at, C, the piston has evidently moved a distance, A D, and the speed of the piston is always that of the point, D.

When a point, such as D, moves on a diameter so as to always be at the foot of the ordinate drawn through a uniformly revolving point as, C, D, is said to move *harmonically* with, C.

We have to find what force must act on the piston at each moment to make it move thus harmonically.

Now a heavy point, C, moving uniformly in a circle, may be supposed to have its motion compounded of a harmonic motion in the direction, A B, and a harmonic motion in the direction, E F, at right angles to, A B. Each of the forces producing these motions taking effect in its own direction, irrespective of forces and motions at right angles to it.

That is, if, D, be a point of equal weight with, C, the force moving, C, horizontally at any moment is the same as that moving, D, because the horizontal velocity of, C, is always equal to that of, D. The force acting vertically on, C, at any moment makes no difference horizontally.

But the resultant of the vertical and horizontal forces on, C, is the force which compels it to move in a circle, that is, it is equal and opposite to what is commonly termed the *centrifugal force* of, C.

* See Paper by the late Prof Fleeming Jenkin, published in the *Transactions of the Royal Society of Edinburgh*, vol. xxviii (1879), p 703, in which will be found a rigorous analysis of the forces in a reciprocating steam engine of practical proportions, accompanied by curves of crank-pin effort.

Therefore the force on the piston making it move harmonically on, A B, is *the horizontal component of the centrifugal force the piston would exert if its weight were concentrated at the centre of the crank-pin and revolved with it.*

If, C O, represents this centrifugal force, then, D O, represents the accelerating force when the piston is at, D.

At the beginning of the stroke, D O = A O. ∴ Erect the perpendicular, A M = A O, to represent the accelerating force when the piston is at, A.

At half stroke, D O = zero.

At any intermediate point, the ordinate, D P, intersected by the straight line, M O, equals, D O, the accelerating force for the position, D.

From half-stroke to the end, the acceleration is negative, and the ordinates must be measured below the line of abscissae The triangle O B N, equal to, O A M, represents the retarding forces.

The algebraic sum of the two triangles, on the principle of the conservation of energy is zero, because the mass starts from rest and comes to rest again. So the inertia merely affects the *distribution* of power during the stroke, not its amount.

We see that the accelerating force is greatest at the ends of the stroke where the motion is slowest, and is *nil* at half stroke where the motion of the piston is fastest. It is not the velocity, but the *rate of change of velocity* which demands the accelerating force.

We have to find the numerical value of the centrifugal force.

This is given in all elementary treatises on mechanics as—

$$\frac{W v^2}{g r} = \frac{m v^2}{r}.$$

Where, v, is velocity in feet per second, r, is radius of circle in feet, and, m, mass in units of mass.

Engineers do not use this notation, for they speak of so many revolutions per minute in a circle of so many feet radius, and they measure mass by weight.

So $m = \frac{W}{g} = \frac{W}{32}$; W being in lbs., and g the acceleration of gravity.

Let N = number of revolutions per minute.

Then,
$$\frac{m v^2}{r} = \frac{W}{32 r} \cdot \left(\frac{2 \pi r N}{60}\right)^2$$

$$,, \quad = \cdot 000341 \, W r N^2.$$

This is a well-known formula for the centrifugal force of a body.

We are now in a position to correct the indicator diagram for any engine, and to say how much less the pressure on the crank-pin is, than that on the piston at the beginning, and how much greater at the end of the stroke.

EXAMPLE I.—The stroke of an engine with a slotted crosshead is 4 feet, the diameter of the cylinder is 20 inches. The effective pressure is 40 lbs. at the beginning and 20 lbs. at the end of the stroke, the weight of the reciprocating mass is 1,256 lbs. What are the pressures on the crank-pin at the beinning and end of the stroke, at 50 revolutions per minute?

Area of a cylinder 20 inches diameter = 314 square inches.

Accelerating force at ends of stroke from the previous formula.

$$\cdot000341\ W r N^2 = \cdot000341 \times 1{,}256 \times 2 \times 2{,}500 = 2{,}141 \text{ lbs.}$$

Pressure on piston at beginning of stroke = 314 × 40 = 12,560 lbs.
Pressure on crank-pin at beginning of stroke 12,560 − 2,141 = 10,419 lbs.
Pressure on piston at end of stroke = 314 × 20 = 6,280 lbs.
Pressure on crank-pin at end of stroke = 6,280 + 2,141 = 8,421 lbs.

The pressure on the crank pin is thus much more equable than would be the case if the parts were not possessed of inertia.

Suppose it is required to make the pressure at the beginning exactly equal to that at the end of the stroke, the weight of the parts being unaltered, but the speed changed What number of revolutions will make the accelerating force at beginning and end of stroke, equal to half the difference of the greatest and least pressures on the piston?

Greatest pressure, . . .	=	12,560 lbs.
Least, ...	=	6,280 lbs.
Half difference, . .	=	3,140 lbs.

Let, x, be the number of revolutions required.

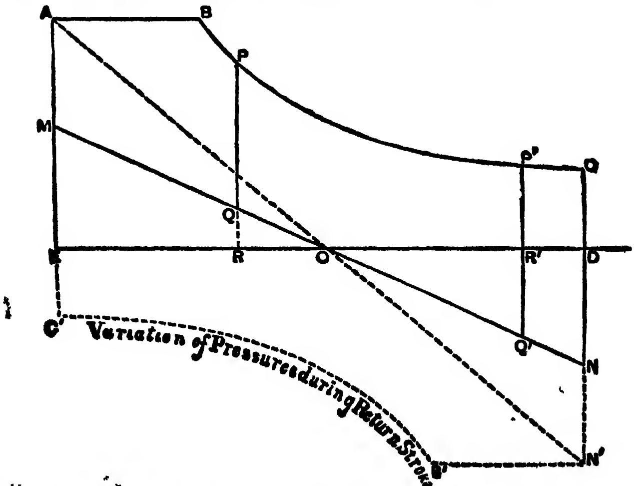
Then, $\cdot000341 \times 1{,}256 \times 2 \times x^2 = 3{,}140$, whence $x = 60$ revs, nearly.

Suppose we are restricted to 50 revolutions, but may vary the weight of the reciprocating parts to obtain the same result.

Let, W, be the weight required.

Then, $\cdot000341 \times W \times 2 \times 2{,}500 = 3{,}140$, ∴ W = 1,840 lbs.

Indicator Diagrams as Modified by Inertia.—To find the pressure as modified by the inertia for *any* point of the stroke, take, A B C D E as the indicator diagram of an engine.



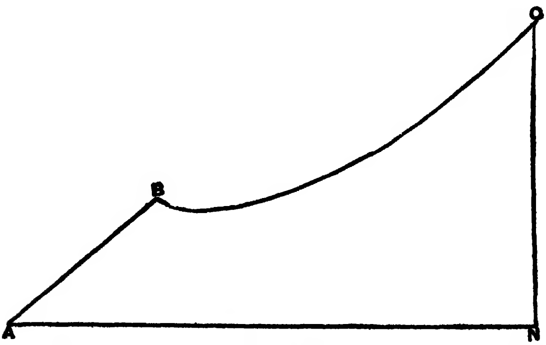
THEORETICAL INDICATOR DIAGRAM CONVERTED INTO A STRESS DIAGRAM ON CRANK-PIN WITH CONNECTING-ROD OF INFINITE LENGTH.

Then to the same scale to which, A E, represents initial steam pressure × area of piston, draw, E M and D N, to represent accelerating force at ends of stroke. Join M and N by the straight line M O N.

Then, M E O D N, is the inertia diagram, and, M O N, thus becomes the virtual base of the figure, from which pressures are to be measured instead of E D. With these heights measured from a horizontal base line the indicator diagram takes the following form :—

PREVIOUS INDICATOR DIAGRAM CORRECTED FOR INERTIA.

Suppose by increasing the speed of the engine or the weight of the reciprocating parts, the ordinate, E M, becomes equal to, A E, that is, the accelerating force is equal to the whole pressure of the steam on the piston: then there is no pressure on the crank-pin when the engine is on the centre, and as the piston advances the pressure will gradually increase, and become excessive towards the end, even with such a comparatively early cut off as shown on the diagram in question. The corrected card is as follows :—



PREVIOUS INDICATOR DIAGRAM CORRECTED FOR INCREASED INERTIA.

With very high speeds or heavy parts the ordinate, E M, may be greater than, E A. The piston will then at the beginning of the stroke lag behind the crank, and be dragged until the acceleration ordinate and steam ordinate become equal. Indeed, with an early cut-off the piston may drag again at a later period of the stroke, as shown in the next figure.

Here the inertia line, M O N, cuts the steam line at, K, L, and R. From, M to K, the pressure is negative, and power represented by area,

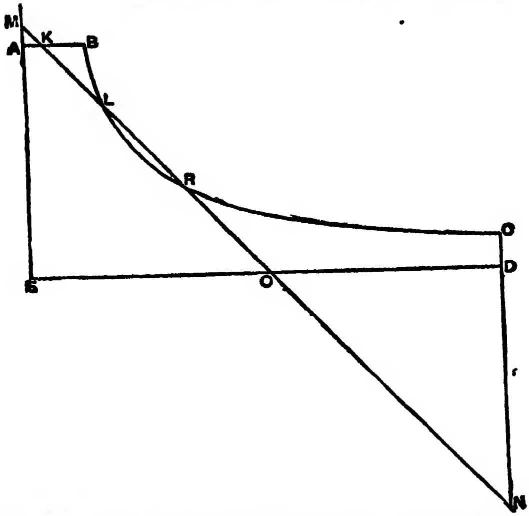
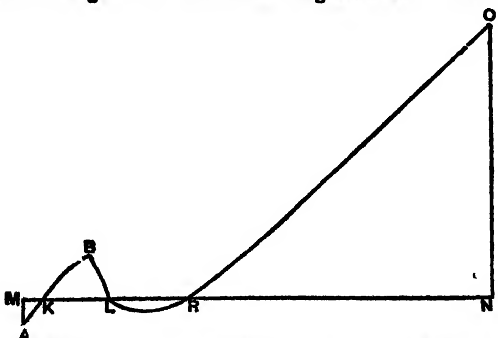


DIAGRAM SHOWING FOUR SHOCKS IN ONE STROKE OF PISTON.

M A K, must be spent on the piston From, K to L, the piston urges the engine, doing work on the crank equal to, K B L. From, L to R, the piston again drags and absorbs work equal to the loop, L R; and thereafter the piston drives the engine. The corrected diagram is as follows :—



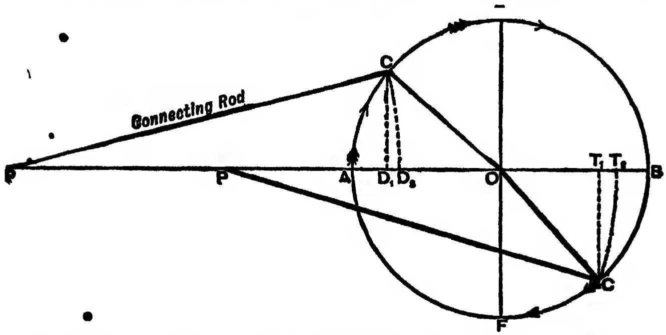
CORRECTED DIAGRAM FOR FOUR SHOCKS IN ONE STROKE.

It is needless to say that such a condition is not desirable in practice.

We now pass on to the more usual case of engines working with a connecting-rod of finite length.

Connecting-rod of Finite Length.—Let, P C, be the connecting-rod and, C O, the crank.

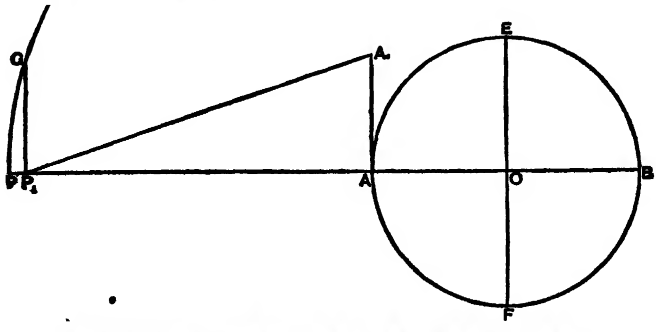
Then at the commencement of the out-stroke from, A to B, the motion of the piston is more rapid than in the pure harmonic motion, the travel being for a certain position of crank-pin, $A D_2$, instead of, $A D_1$. At the beginning of the return stroke from, B to A, the speed is less than for harmonic motion, the travel corresponding to the position of crank-pin shown being, $B T_2$, instead of, $B T_1$.



MOTIONS OF PISTON WITH A CONNECTING-ROD OF FINITE LENGTH.

It is plain that the mere raising or lowering the big end of the connecting-rod in a vertical line, will make, P, approach, O, with a certain acceleration. And the force necessary for this, must be added to that required for true harmonic motion when the engine is on the 'near' centre, A, and subtracted for the 'far' centre, B.

Now, what is this accelerating force?



ACCELERATING FORCE AT INNER DEAD CENTRE DUE TO A CONNECTING-ROD OF FINITE LENGTH.

In the above fig., P A, is the connecting-rod as before. We suppose the end, A, to ascend the vertical line, $A A_1$; then, P, will be drawn to, P_1, and, $P_1 A_1$, is a position of the connecting-rod.

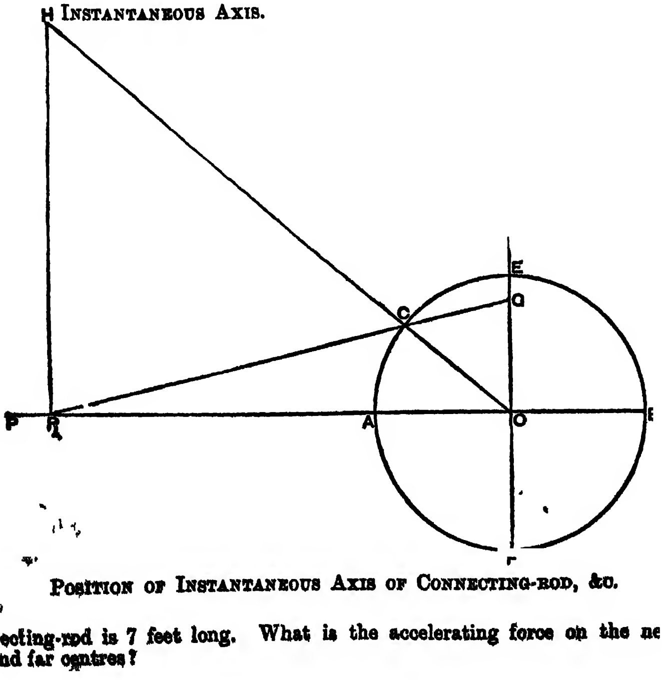
With centre, A, describe the arc, P G. Now we are only concerned with the velocity of, A, *at the moment it leaves the line, P O:* what its velocity is afterwards cannot affect the acceleration of, P, at that moment.

Suppose, therefore, the point, A, to ascend, A A_1, harmonically with the point, G, revolving uniformly in the circle, P G, with the velocity of the crank-pin. Then, obviously, P_1, moves also harmonically with, G, on line, P O.

Therefore, when the big end of the connecting-rod leaves the centre line, P O, with the said velocity, the accelerating force on, P, is the centrifugal force it would have when moving in a circle with radius equal to the connecting-rod, with the velocity of the crank-pin.

But with a given linear velocity, the centrifugal force is inversely as the radius: therefore, if the connecting-rod is, n, times the length of the crank, the accelerating force due to the connecting-rod will be, $\frac{1}{n}$*th*, that due to the crank, and the net accelerating force when the engine is on the near centre will be, $1+\frac{1}{n}$, and when on the far centre, $1-\frac{1}{n}$, times that of an engine with pure harmonic motion.

EXAMPLE II.—The reciprocating parts of an engine weigh one ton. The stroke is 3 feet 6 inches, and the revolutions 80 per minute. The con-



POSITION OF INSTANTANEOUS AXIS OF CONNECTING-ROD, &c.

necting-rod is 7 feet long. What is the accelerating force on the ne and far centres?

Here $n = 4$. Therefore the accelerating force on near centre will be—
$\cdot000341\ W\,r\,N^2$.

$$= \cdot000341 \times 2{,}240 \times 1{\cdot}75 \times 6{,}400 \times \frac{5}{4} = 10{,}700 \text{ lbs.}$$

And on far centre $10{,}700 \times \frac{3}{5} = 6{,}420$ lbs.

It is evident that the accelerating forces, when the engine is on the "dead points," are by far the most important; first, because they are there greatest, and secondly, because the motion of the piston changes its direction there.

We may, however, investigate one other point in the inertia diagram, viz.:—that at which the acceleration is *nil*, or the point where the line corresponding to, M O N, in the figures for an infinite connecting-rod, cuts the base line. (See fig. on previous page).

Obviously the acceleration is *nil* when the speed of the piston is greatest.

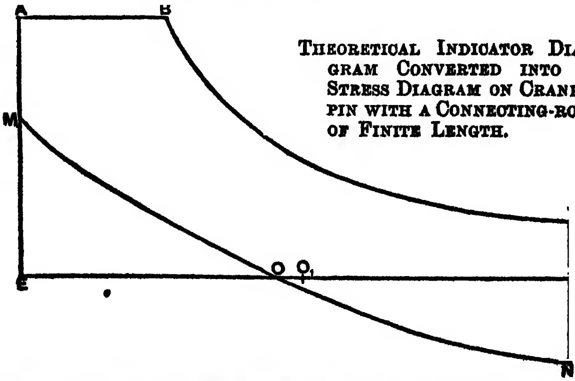
Let, P_1 C, be any position of the connecting-rod. Produce, P_1 C, to cut O E, in, G. Draw, P_1 H, at right angles to, P O, and produce, O C, to cut, P_1 H, in, H. (See fig. on previous page).

The connecting-rod at any moment is moving about an *instantaneous axis:* and every point in it is, of course, moving at right angles to the line joining it to this axis. The axis is, therefore, somewhere in the line, P_1 H, and also somewhere in the line, O C H, for, C, is moving at right angles to, O C.

Therefore, H, *is the instantaneous axis*, and the velocities of, P_1 and C, are in the ratio, H P_1 to H C. That is, by similar triangles, the ratio, O G to O C. O G, therefore represents *the velocity of the piston.*

By drawing the length of the connecting rod on a piece of tracing paper and applying it to the diagram, keeping, P_1, in the line, P O, and, C, in the circumference, a position of, P_1, can be found for which, O G, is a maximum. To find this position of, P, by calculation requires the use of the higher mathematics, which is purposely avoided here.

Take the indicator diagram of an engine as, A B C D E.



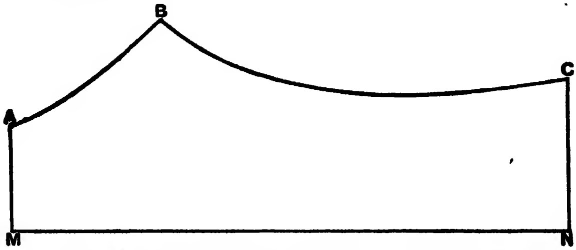
THEORETICAL INDICATOR DIAGRAM CONVERTED INTO A STRESS DIAGRAM ON CRANK-PIN WITH A CONNECTING-ROD OF FINITE LENGTH.

Suppose the connecting-rod is 4 cranks long, a very common proportion. Find, by trial, with a piece of tracing paper, as above, the position, O, of the piston when its velocity is greatest. With this proportion o

connecting-rod, O, will be found at a distance, OO_1, from mid-stroke a little more than $\frac{1}{10}$ of the half stroke, O_1E. Mark off to the proper scale, $EM = 1\frac{1}{4}$ times the centrifugal force of a weight equal to that of reciprocating parts revolving with the crank-pin, and, $DN = \frac{3}{4}$ of that force.

(If the connecting-rod were 5 cranks long these amounts would be respectively $1\frac{1}{5}$ and $\frac{4}{5}$ of the force).

Draw a fair curve through, MON, bearing in mind that the triangles, MOE, ODN, must be equal in area, representing as they do the same, *vis vira*. Though not mathematically exact, this line, MON, will very approximately represent the inertia diagram: and the pressures on the crank-pin are measured by vertical lengths, intercepted between, ABC and MON. Placed on a horizontal base, the amended diagram becomes the following figure.



CORRECTED FIGURE FOR THE PREVIOUS DIAGRAM.

It will be seen that a considerable error in the line, MON, will not greatly affect the shape of the figure, provided the points, M and N, are accurately determined by the method given above. The point, O, has been referred to, not because it is important, but in order to show the student that it is unimportant.

We have hitherto supposed all the reciprocating weights as having the same motion as the piston, and as being concentrated at, P or P_1, the centre of the crosshead. This supposition introduces no error into the estimation of accelerating forces caused by the motion of the crank alone: but in the case of the connecting-rod it will be seen that the motion of its *centre of gravity* caused by moving the "big end" off the centre line (see lower fig. on p. 297), is not the motion of, P_1, towards, A, but one-half that amount, supposing the centre of gravity to be at the middle of the rod. Therefore, strictly, if, W, be the weight of all purely reciprocating parts, such as piston, piston-rod, and crosshead, and, *w*, the weight of the connecting-rod, the accelerating force on the centre instead of being

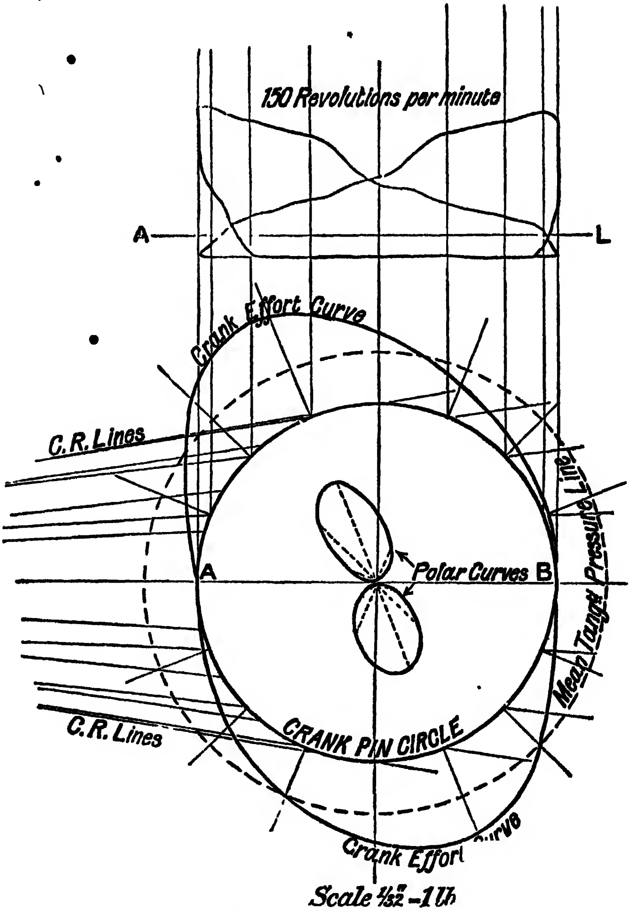
$$\cdot000341\,(W + w)\,r\,N^2\left(1 \pm \frac{1}{n}\right)$$

is,
$$\cdot000341\,W\,r\,N^2\left(1 \pm \frac{1}{n}\right) + \cdot000341\,w\,r\,N^2\left(1 \pm \frac{1}{2n}\right).$$

If instead of being horizontal the engine be vertical, the effect of inertia is just the same, but the pressure on the crank-pin will be further affected by the weight of the reciprocating parts, which must be added at every point of the down stroke, and subtracted for the up stroke.

In the case of a diagonal engine, the vertical component of the weight must be so added or subtracted.

Crank Effort Diagrams.—We shall now explain, by aid of the following figure, how to construct a crank effort diagram, when the obliquity of the connecting-rod and the varying pressures on the piston are taken into account.*



CRANK EFFORT CURVES OF "THE THOMAS RUSSELL ENGINE."

* The front and back indicator diagrams (from which the above crank effort curve has been drawn) were taken from "The Thomas Russell Experimental Steam Engine" in the author's laboratory. The diameter of the cylinder = 6 ins.; length of stroke = 12 ins.; length of connecting-rod = 36 ins.; revolutions per minute = 150.

(1) Transfer the indicator diagrams from the cards to a sheet of paper, taking care to make them at least 4 inches in length for clearness.

(2) Some distance below the redrawn diagrams put down a line, A B, parallel to the atmospheric line and equal in length to the diagrams.

(3) Upon A B, as a diameter, draw the crank-pin circle and divide the same into any convenient number of equal parts.

(4) Project each of these points of division vertically upwards, so as to cut the indicator diagrams.

(5) For the first point to be considered on the crank-pin circle, take that which is vertically over the centre of the crank. Measure the pressure on the piston (from the indicator diagram) corresponding to this point and plot it along the centre line of the connecting-rod as produced through this point.

(6) Resolve this pressure (as plotted along the centre line of the connecting-rod) into two forces at right angles to each other—viz., one along the crank centre line and the other at right angles to it—*i.e.*, tangentially to the crank pin circle.

(7) Do precisely the same for each of the other positions into which the crank-pin circle is divided.

(8) Produce the several centre lines of the crank for each of these positions and plot off on each centre line, from the crank-pin circle, the corresponding tangential pressure to the same scale as the indicated diagrams. By joining these points the crank effort diagram is obtained.

(9) *To Find the Mean Tangential Pressure Line—*

Let P_m = Mean pressure on piston (from indicator diagrams).

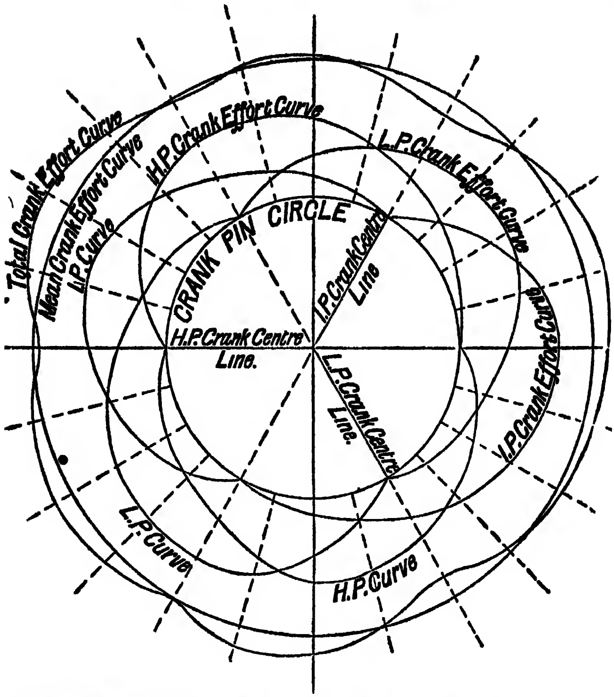
" r = Radius of crank.

" P_t = Mean tangential pressure on crank-pin.

Then
$$P_m \times 2r = P_t \times \pi r$$

$$\therefore \quad P_t = \frac{P_m \times 2r}{\pi r} = \frac{2P_m}{\pi} = \frac{P_m}{1{\cdot}57}.$$

(10) In drawing the "crank effort curves" for compound, triple, or quadruple expansion engines, the several indicator diagrams have to be reduced to one scale, as explained in Lecture XVI. Then, the several steam pressures (for the several positions assumed on the crank-pin circle), *in the case of the intermediate and low-pressure cylinders*, must be multiplied by the ratios of the areas of those cylinders with respect to the area of the high-pressure cylinder *before* resolving their respective piston pressures along the centre lines of the several positions occupied by their connecting-rods. We have been fortunate in securing, for a practical example with which to illustrate this case, copies of the indicator diagrams taken on the trial trip of a steamer recently built by one of the best Clyde firms. There was nothing abnormal about the diagrams; consequently we need not reproduce them on the combined diagrams to one scale, from which we obtained the following set of "crank effort curves." The "total crank effort curve" is obtained by summing up the pressures on the three-crank effort curves along the respective radial ordinates to the tangential pressures on the crank-pin circle. The "mean crank effort curve" is found, as explained under (9). The chief data connected with this case is given immediately below the figure.



CRANK EFFORT DIAGRAM OF A TRIPLE EXPANSION ENGINE.

Ratio of expansion, 10 4

Length of connecting-rod = 9 feet. Stroke = 4·5 feet.

Mean efficiency of steam, or ratio of area of work in cylinder to full theoretical diagram, 55 per cent.

	H P	I P.	L P
Cylinder's diameter, ...	28″	46″	77″
Area, ...	615·8	1661·9	4656·6
Ratio, ...	1	2 80	7·56
Mean pressures, lbs. per sq. in., .	67·6	28·2	9·7
Range of temperatures, Fah., .	64·3°	74·9°	80·6°

Steam, 164 lbs.; Vacuum, 26½ ins.; Receivers, 52 and 5 lbs.; Revolutions, 62¼ per minute.

Cut-off H.P. = 33¼ ins.	I.H.P. = 710 L.P.
,, I.P. = ,,	,, = 799 I.P.
,, L.P. = ,,	,, = 764 H.P.

2273 total I.H.P.

Crank Effort Diagram of the Quadruple-Expansion Five-Crank Engines of S.S. "Inchdune."—The illustrated description of these engines is given in Lecture XXIII. The necessary calculations with explanations and the drawings from which the accompanying folding-plate has been directly reproduced by photography were kindly supplied to the author by his old student, Mr. William C. Borrowman, M.Inst.C.E., General Manager of the Central Marine Engineering Works, West Hartlepool, who designed the arrangement of engines and boilers.

The following calculations show clearly how the foregoing educational illustrations and descriptions are applied in practice. The results thus obtained were highly gratifying to the designer, not only in the even subdivision of the powers derived from the five cylinders, but in the mean crank effort diagram, comparative freedom from stresses and vibration; but also in the unprecedented economy in steam and coal of these engines and boilers. (*See the facing folding-plate.*)

S.S. "INCHDUNE" ENGINES.

Preliminary Data.—

Diameter of cylinders, . .	17, 24, 34, 42, 42 inches.
Length of stroke, ...	42 ,,
Diameter of piston-rods, ...	$4\frac{1}{2}$,,
Length of connecting-rod, ...	$7\frac{1}{2}$ feet.
Boiler pressure = 265 lbs. by gauge, or 280 lbs absolute.	
Vacuum by gauge, ...	= 27 inches.

Method of Calculating Inertia and Gravitating Effect of the Reciprocating Parts—viz., piston, piston-rod and $\frac{1}{2}$ connecting-rod. (The crank-pin, webs, and $\frac{1}{2}$ connecting-rod are supposed to balance each other.)

$$\left.\begin{array}{r}\text{Inertia effect for an infinite}\\ \text{connecting-rod}\end{array}\right\} = \frac{Wv^2}{gr} = \frac{v^2}{g} = \frac{(2\pi \times N)^2}{(60)^2 \times 32{\cdot}2}$$

$$\text{,,} \qquad \text{,,} \qquad = \frac{(2\pi N)^2}{32{\cdot}2 \times 60^2} = \frac{4 \times 22 \times 22 \times 100^2}{7 \times 7 \times 32{\cdot}2 \times 60^2}$$

$$\text{,,} \qquad \text{,,} \qquad = 3{\cdot}4056.$$

When W = 1 ton (weight of reciprocating parts).
r = 1 foot = radius of crank-pin circle.
N = 100 revolutions per minute.
v = velocity of crank-pin in feet per second.

$$\left.\begin{array}{r}\text{Correction of inertia}\\ \text{effect for finite rod}\end{array}\right\} = \left\{\begin{array}{l}\text{Inertia effect for an infinite con-}\\ \text{necting-rod} \times \text{constant.}\end{array}\right.$$

$$\text{,,} \quad \text{,,} \quad = 3{\cdot}4056 \times \cos\theta + \frac{n^2 \cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}} = \pm f.$$

When θ = angle made by crank with line of stroke.

l = length of connecting feet in feet.

r = length of crank in feet.

$$n = \frac{\text{length of connecting rod}}{\text{length of crank}} = \frac{l}{r}.$$

But, as inertia effect increases directly as W, r, and N^2,

$$\therefore \quad \pm f\left(W \times r \times \frac{N^2}{(100)^2}\right) = \pm f \times \text{a constant.}$$

$$\text{In this case, } n = \frac{\text{length of connecting-rod}}{\text{length of crank}} = \frac{l}{r} = \frac{7{\cdot}5}{1{\cdot}75} = 4{\cdot}285.$$

N = revolutions per minute = 70.

l = length of connecting-rod = 7·5 feet.

r = radius of crank = 1·75 feet.

$$\text{Then,} \quad \text{Inertia effect} = \pm f\left(Wr\frac{N^2}{(100)^2}\right)$$

$$\text{,,} \quad \text{,,} \quad = \pm f\left(W \times 1{\cdot}75 \times \frac{70 \times 70}{100 \times 100}\right).$$

$$\left.\begin{array}{l}\text{Hence, Total effect of}\\ \text{inertia and gravita-}\\ \text{tion, F}\end{array}\right\} = \pm f\left(W \times 1{\cdot}75 \times \frac{70 \times 70}{100 \times 100}\right) + W.$$

(For down stroke.)

$$\text{,,} \quad \text{,,} \quad \text{,,} \quad = \mp f\left(W \times 1{\cdot}75 \times \frac{70 \times 70}{100 \times 100}\right) - W.$$

(For up stroke.)

Cylinder Number	Weight of Reciprocating Parts in Tons	Total Inertia and Gravitation Effect
1	9569	$\pm (f \times \cdot 8205) \pm \cdot 957$
2	1·108	$\pm (f \times \cdot 9501) \pm 1{\cdot}108$
3	1 437	$\pm (f \times 1{\cdot}2327) \pm 1\ 437$
4F	1·4634	$\pm (f \times 1{\cdot}2549) \pm 1{\cdot}463$
4A	1·59	$\pm (f \times 1{\cdot}3634) \pm 1\ 590$

Crank Effort Diagrams for No. 1 Cylinder of S.S. "Inchdune" during the Down Stroke.

Crank Angle in Degs.	Crank Leverage in Inches.	Steam Pressure in Lbs. per Sq. Inch Absolute.	Total Pressure on Piston in Tons.	f		Constant.		Inertia in Tons.		Weight of Reciprocating Parts, W, in Tons.		Gravity and Inertia Effect, F, due to Reciprocating Parts, in Tons.*	Turning Moment, T.M., in Inch-Tons, due to Gravity and Inertia only.	Turning Moment, T.M., in Inch-Tons, due to Steam only.	Total Turning Moment, T.M., in Inch-Tons, due to Gravity, Inertia and Steam.
0	0	147	14·890	−4·200	×	·8205	=	−3·446	+	·957	=	−2·489	0	0	0
10	4·48	146	14·790	−4·102	×	·8205	=	−3·366	+	·957	=	−2·409	−10·792	66·258	55·466
20	8·74	140	14·182	−3·815	×	·8205	=	−3·130	+	·957	=	−2·173	−18·992	123·951	104·959
30	12·61	134	13·574	−3·358	×	·8205	=	−2·755	+	·957	=	−1·798	−22·672	171·171	148·499
40	15·94	127	12·865	−2·759	×	·8205	=	−2·264	+	·957	=	−1·307	−20·833	205·069	184·236
50	18·52	122	12·359	−2·060	×	·8205	=	−1·690	+	·957	=	− ·733	−13·575	228·881	215·306
60	20·34	118	11·953	−1·299	×	·8205	=	−1·066	+	·957	=	− ·109	− 2·217	243·132	240·915
70	21·33	116	11·751	− ·546	×	·8205	=	− ·448	+	·957	=	+ ·509	+10·857	250·645	261·502
80	21·52	111	11·244	+ ·175	×	·8205	=	+ ·144	+	·957	=	+1·101	+23·693	241·977	265·670
90	21·00	103	10·434	+ ·817	×	·8205	=	+ ·670	+	·957	=	+1·627	+34·167	219·112	253·279
100	19·80	80	8·104	+1·357	×	·8205	=	+1·113	+	·957	=	+2·070	+40·986	160·459	201·445
110	18·11	61	6·179	+1·784	×	·8205	=	+1·464	+	·957	=	+2·421	+43·844	111·907	155·751
120	16·01	37	3·748	+2·107	×	·8205	=	+1·729	+	·957	=	+2·686	+43·003	60·007	103·010
130	13·82	− 8	− ·810	+2·318	×	·8205	=	+1·902	+	·957	=	+2·859	+39·511	−11·200	28·311
140	11·05	− 48	− 4·862	+2·458	×	·8205	=	+2·017	+	·957	=	+2·974	+32·862	−53·729	−20·867
150	8·36	− 90	− 9·117	+2·541	×	·8205	=	+2·085	+	·957	=	+3·042	+25·431	−76·218	−50·787
160	5·52	−120	−12·156	+2·585	×	·8205	=	+2·121	+	·957	=	+3·078	+16·990	−67·101	−50·111
170	2·92	−131	−13·270	+2·605	×	·8205	=	+2·137	+	·957	=	+3·094	+ 9·035	−38·749	−29·714
180	0	−140	−14·182	+2·611	×	·8205	=	+2·142	+	·957	=	+3·099	0	0	0
															2,066·87

* It will be seen for this column that, for the same position of the piston, the combined inertia and gravitation effect for the *up stroke* is the same as for the *down stroke*, but with the signs changed.

Crank Effort Diagrams for No. 1 Cylinder of S.S. "Inchdune" during the Up Stroke.

Crank Angle in Degs.	Crank Leverage in Inches.	Steam Pressure in Lbs. per Sq. Inch Absolute.	Total Pressure on Piston in Tons.	f		Constant.		Inertia in Tons.		Weight of Reciprocating Parts, W, in Tons.		Gravity and Inertia Effect, F, due to Reciprocating Parts, in Tons.*	Turning Moment, T M, in Inch-Tons, due to Gravity and Inertia only.	Turning Moment, T M, in Inch-Tons, due to Steam only.	Total Turning Moment, T M, in Inch-Tons, due to Gravity, Inertia and Steam.
180	0	140	13·188	−2·611	×	·8205	=	−2·142	−	·957	=	−3·099	0	0	0
190	2·92	140	13·188	−2·605	×	·8205	=	−2·137	−	·957	=	−3·094	−9·035	38·509	29·474
200	2·52	140	13·188	−2·585	×	·8205	=	−2·121	−	·957	=	−3·078	−16·990	72·800	55·810
210	8·36	139	13·094	−2·541	×	·8205	=	−2·085	−	·957	=	−3·042	−25·431	109·464	84·033
220	11·05	138	13·000	−2·458	×	·8205	=	−2·017	−	·957	=	−2·974	−32·862	143·646	110·784
230	13·82	133	12·529	−2·318	×	·8205	=	−1·902	−	·957	=	−2·859	−39·511	173·144	133·633
240	16·01	129	12·152	−2·107	×	·8205	=	−1·729	−	·957	=	−2·686	−43·003	194·550	151·547
250	18·11	124	11·681	−1·784	×	·8205	=	−1·464	−	·957	=	−2·421	−43·844	211·539	167·695
260	19·80	117	11·021	−1·357	×	·8205	=	−1·113	−	·957	=	−2·070	−40·986	218·224	177·238
270	21·00	112	10·550	−·817	×	·8205	=	−·670	−	·957	=	−1·627	−34·167	221·558	187·391
280	21·52	102	9·608	−·175	×	·8205	=	−·144	−	·957	=	−1·101	−23·693	206·773	183·080
290	21·33	76	7·159	+·546	×	·8205	=	+·448	−	·957	=	−·509	−10·857	152·706	141·849
300	20·34	53	4·993	+1·299	×	·8205	=	+1·066	−	·957	=	+·109	+2·217	101·549	103·766
310	18·52	32	3·014	+2·060	×	·8205	=	+1·690	−	·957	=	+·733	+13·575	55·827	69·402
320	15·94	−13	−1·225	+2·759	×	·8205	=	+2·264	−	·957	=	+1·307	+20·833	−19·520	1·313
330	12·61	−62	−5·840	+3·358	×	·8205	=	+2·755	−	·957	=	+1·798	+22·672	−73·647	−50·975
340	8·74	−122	−11·492	+3·815	×	·8205	=	+3·130	−	·957	=	+2·173	+18·992	−100·444	−81·452
350	4·48	−158	−14·883	+4·102	×	·8205	=	+3·366	−	·957	=	+2·409	+10·792	−66·678	−55·886
360	0	−162	−15·260	+4·200	×	·8205	=	+3·466	−	·957	=	+2·489	0	0	0
															1,408·702

* It will be seen for this column that, for the same position of the piston, the combined inertia and gravitation effect for the *up stroke* is the same as for the *down stroke*, but with the signs changed. The turning moments for each of the other cranks were ascertained in the same way for their down and up strokes. The mean results are given in the next table.

Effective crank leverage calculated from the following formula:—

$$\left.\begin{array}{r}\text{Effective leverage, L, of}\\ \text{crank at angle } \theta\end{array}\right\} = r\sin\theta\left(1 + \frac{r\cos\theta}{\sqrt{l^2 - r^2\sin^2\theta}}\right).$$

When, θ = angle of crank with line of stroke.
r = length of crank in feet.
And l = length of connecting-rod in feet.

Travel of piston, T, from commencement of stroke when crank is at an angle $\theta = r(1 - \cos\theta) \pm l \mp \sqrt{l^2 - r^2\sin^2\theta}$.

Down stroke:—

$$\left.\begin{array}{r}\text{Total inertia and}\\ \text{gravitation effect, F}\end{array}\right\} = \mp f\left(\text{W}r\frac{\text{N}^2}{(100)^2}\right) + \text{W}.$$

$$\text{,,} \quad \text{,,} \quad = \mp f\left(\cdot 9569 \times 1\cdot 75 \times \frac{70 \times 70}{100 \times 100}\right) + \cdot 9569.$$

$$\text{,,} \quad \text{,,} \quad = \mp f \times \cdot 8205 + \cdot 9569.$$

Up stroke:—

$$\left.\begin{array}{r}\text{Total inertia and}\\ \text{gravitation effect, F}\end{array}\right\} = \mp f\left(\text{W}r\frac{\text{N}^2}{(100)^2}\right) - \text{W}.$$

$$\text{,,} \quad \text{,,} \quad = \mp f\left(\cdot 9569 \times 1\cdot 75 \times \frac{70 \times 70}{100 \times 100}\right) - \cdot 9569.$$

$$\text{,,} \quad \text{,,} \quad = \mp f \times \cdot 8205 - \cdot 9569.$$

$$\left.\begin{array}{r}\text{Mean turning moment,}\\ \text{T M, in inch-tons for}\\ \text{No. 1 cylinder}\end{array}\right\} = \frac{\text{Total T M during down stroke} + \text{total T M during up stroke}}{36}.$$

$$\text{,,} \quad \text{,,} \quad = \frac{2066\cdot 87 + 1408\cdot 702}{36}$$

$$= \frac{3475\cdot 572}{36} = 96\cdot 544 \text{ inch-tons.}$$

NOTE.—Piston-rod areas have been deducted in working up the mean pressures of the indicator cards.

$$\left.\begin{array}{r}\text{Mean turning moment,}\\ \text{T M, from I.H.P.}\end{array}\right\} = \frac{\text{I.H.P.} \times 33{,}000 \times 12}{2{,}240 \times 2\pi \times \text{revolutions per min.}}$$

TABLE SHOWING THE I.H.P., MEAN TURNING MOMENTS FOR EACH CRANK, AND THE TOTALS FOR THE S.S. "INCHDUNE."

Cylinder Number	Indicated Horse-Power, I H P.	Mean Turning Moment, T M, from I.H P	Mean Turning Moment, T M, from above Calculation.
1	240	96·516	96 541
2	236	94 907	94 337
3	252	101·342	101·399
4F	263	105·765	106 132
4A	278	111 798	111·866
Total, .	1,269	510 328	510 278

$$\left.\begin{array}{l}\text{Percentage difference between}\\ \text{mean turning moments}\end{array}\right\} = (510{\cdot}328 - 510{\cdot}278) \times \frac{100}{510\ 328}$$

$$\text{,,} \qquad \text{,,} \qquad = {\cdot}009 \text{ per cent. error.}$$

EXAMPLE III.—What is the effect of inertia and gravitation on the reciprocating parts of a vertical engine having a stroke of 42 inches, connecting-rod equal 4 cranks, weight of reciprocating parts 2 tons, position of crank from commencement of down stroke 30°, revolutions per minute 72?

$$F = W\left\{\left(-fr\frac{N^2}{(100)^2}\right) + 1\right\}$$

$$F = 2\left\{\left(-3{\cdot}3876 \times 1{\cdot}75 \times \frac{(72)^2}{(100)^2}\right) + 1\right\} = -4{\cdot}1465 \text{ tons.}$$

Lecture XVIII.—Questions.

1. In a double-acting engine the mean pressure on the piston is 4 tons and the length of the stroke 18 inches, what is the mean pressure which can be taken from the rim of the fly-wheel, the estimated diameter of which is 8 feet? *Ans.* 1069 lbs. (about).

2. The crank of a steam engine is 2 feet long, and the mean tangential force acting upon it is 17,000 lbs., what is the mean pressure of the steam upon the piston of the engine during each stroke? *Ans.* 26703·6 lbs.

3. In a direct-acting engine the diameter of the cylinder is 17 inches, and the mean pressure of the steam 60 lbs., the crank being 12 inches long, what is the mean pressure on crank in the direction of its motion? *Ans.* 8,670 lbs.

4. Explain the manner in which the reciprocating motion of the piston in a locomotive engine is converted into the rotatory motion of the crank shaft. What are the dead points? Show by the principle of work that there is no loss of power by the intervention of the crank, friction being disregarded.

5. Explain the method of representing in a diagram, the work done during one revolution of the crank of an engine by setting off ordinates representing the tangential efforts on the crank pin.

6. In a direct-acting engine the crank and connecting-rod are as 1 to 6. Find an expression for the tangential pressure on the crank pin in any position. Construct an approximate diagram of work done upon the crank during the stroke, and give a sketch of the same, (1) when there is a single cylinder, and (2) when there are two cylinders working cranks at right angles.

7. In a horizontal direct-acting engine you are required to find an expression for the tangential force upon the crank pin in any given position of the crank. Example—The lengths of the crank and connecting-rod being 1 and 6 respectively, and the pressure on the steam piston being 2,000 lbs., estimate the tangential force on the crank when in a vertical position. Find also the vertical force acting upon the crank shaft. *Ans.* 2,000 lbs.; 333 lbs.

8. In a direct-acting horizontal engine the length of the crank 1 foot and that of the connecting-rod is 5 feet. When the crank is vertical the pressure of the steam on the piston is 4,000 lbs.; find the thrust along the connecting-rod, and the pressure on the guide bars at that point of the stroke. *Ans.* 4082·5 lbs.; 816·5 lbs.

9. If the cylinder of a locomotive be 20 inches in diameter with a stroke of 2 feet, and the diameter of the driving wheel be 6 feet, find the tractive force exerted by the engine for each pound of pressure per square inch on the piston. *Ans.* 66·6 lbs.

10. Explain the effects of the inertia of the reciprocating parts in a reciprocating engine, and taking a particular case, work out a crank-pin stress diagram.

11. Draw an indicator diagram of a corliss (or some engine with instantaneous cut-off), in which the cut-off took place at ¼ stroke. From this construct a diagram of crank effort (1) for a single cylinder engine, (2) for double cylinder engine with cranks at right angles.

12. In a direct-acting engine, find the ratio of the velocity of the crank-pin to that of the piston in any given position of the crank.

13. The crank of an engine has a radius of 18 inches, the connecting-rod is 6 feet long, and the number of revolutions made by the engine is 80 per minute. Find graphically or otherwise the velocity of the piston in feet per second when the crank has passed through an angle of 30° from the dead centre during the forward stroke. *Ans.* 7·65 feet per second.

14. In a direct-acting horizontal engine, where the connecting-rod works between guides, the connecting-rod is five times as long as the crank, the pressure on the piston when the crank is vertical being 1,250 lbs.; find the thrust on the slide bar, neglecting friction, and indicate the direction in which it acts. Does the direction of the thrust change during any part of the revolution? *Ans.* 255 lbs.

15. In a direct-acting engine the crank is 2 feet in length, and the connecting-rod is 8 feet; find the distance in inches of the piston from the middle point of its stroke, when the crank is at 90° from a dead centre. Answer this by calculation as well as graphically. *Ans.* 3·05 inches.

16. A steam engine with a cylinder of D inches in diameter, receives steam at 80 lbs. absolute pressure per square inch, and the cut-off is at $\frac{7}{8}$ of the stroke. Find an expression for the diameter of the cylinder of another engine with the same stroke and piston speed which develops the same horse-power as the first engine, but which cuts off the steam at $\frac{1}{4}$ stroke. What would be the relative maximum stresses on the crank-pin and crank-shaft of the two engines when both transmit the same power, the inertia of the reciprocating parts and the obliquity of the connecting-rod being neglected?

17. What is the effect of inertia and gravitation on the reciprocating parts of a vertical engine having a stroke of 60 inches; connecting-rod equal four cranks; weight of reciprocating parts, 3 tons; position of crank from commencement of up stroke, 30°; revolutions per minute, 150? Plot the inertia and leverage diagram for this engine, making the radius of the diagram equal by scale to the calculated centrifugal force or inertia effect for an infinite connecting-rod acting at the commencement of stroke upon the crank-pin.

$$F = W\left\{\left(-fr\frac{N^2}{100^2}\right) - 1\right\} = 3\left\{\left(-2\cdot5099 \times 2\cdot5 \times \frac{150^2}{100^2}\right) - 1\right\}$$

$$F = -45\cdot354 \text{ tons.}$$

18. Describe and show graphically how the force transmitted to the crank-pin is affected by the inertia of the moving reciprocating parts in a stationary horizontal engine during the forward and backward strokes of the piston respectively. How is the force affected by an increase in the ratio of expansion of the steam, by the shortness of the connecting-rod, and by the momentum of the flywheel? In a horizontal non-condensing engine, whose cylinder is 16 inches diameter, stroke 28 inches, connecting-rod 5 feet 10 inches in length, making 100 revolutions per minute, and working with steam whose initial gauge pressure is 140 lbs. per square inch, when cutting-off takes place at four-tenths of the stroke, what would be the pressure on the crosshead at the beginning and at the end of the stroke if the weight of the reciprocating parts is 350 lbs.? and show, by a diagram, the variation in the pressure as the piston makes its forward stroke, the effect of the cushioning of the steam at the end of the stroke being neglected; expansion as if pv were constant. (S. & A., 1897, Hons.)

19. The piston and all that is rigidly connected with it weigh 500 lbs., the crank is 1 foot long, and the speed 200 revolutions per minute. Show, on a diagram, the forces which must be exerted at the crosshead during a

revolution if there is no steam pressure. Choose a pair of indicator diagrams, and show how we find the diagram of forces acting at the crosshead. Assume an infinitely long connecting-rod. (S. & A., 1898, H., Part i.)

20. If, on a piston of 120 square inches in area and weighing with piston-rod 290 lbs., there is at a certain instant a pressure of 130 lbs. per square inch on one side more than what there is on the other, and if the piston acceleration at that instant is 420 feet per second per second in the direction in which the steam is urging the piston, what is the total force acting at the crosshead? If this acceleration occurs when the piston is one-quarter of its stroke from one end, assuming an infinitely long connecting-rod, how many revolutions per minute is the engine making? The crank is 1 foot long. (B. of E., 1900, Adv. and H., Part i.)

21. Explain fully the influence of the reciprocating parts in modifying the effective steam pressure in an engine, and show how to make the calculations necessary in order to determine the weight of flywheel needed to keep the fluctuation of speed during one revolution of an engine within any chosen limit. (C. & G., 1900, H., Sec. B.)

22. If the connecting-rod is 5 feet long, and the crank is 1 foot; 200 revolutions per minute; what are the accelerations of the piston when it is farthest from and nearest to the crank? The piston and rod and crosshead weigh 330 lbs. Area of piston 120 square inches. At the beginning of either the in or out stroke the pressure is 80 lbs. per square inch on one side in excess of what it is on the other. Find the total forces on the crosshead in these two cases. (B. of E., 1902, H., Part i.)

23. Sketch a real probable indicator diagram for a non-condensing engine, single cylinder, 18 inches diameter, crank 15 inches, 150 revolutions per minute, cutting off at about half-stroke, with slide-valve. Neglecting inertia effects, show how we find the turning moment on the crank-shaft in every position, and how it usually varies. What effect has this change of moment upon the strength of the crank-shaft? (B. of E., 1902, H., Part i.)

24. Piston 115 square inches in area. At the beginning of either stroke there is a difference of pressure of 90 lbs. per square inch on its two sides, producing total force in the direction in which the piston is about to move. The piston and its rod weigh 410 lbs. The engine makes 130 revolutions per minute; crank 1 foot. Neglecting angularity of connecting-rod—that is, assuming that the piston has a simple harmonic motion—what is the actual force at the crosshead at the beginning of either stroke? What correction must be made when the angularity of the connecting-rod is not neglected; (B. of E., 1903, Adv. and H., Part i.)

25. Explain what is meant by the "pressure due to the inertia of the reciprocating parts" in a steam engine, and show how it modifies the effective crank effort at different points of the stroke. If the revolutions per minute are 400 and the mean piston speed 1,200 feet per minute, show that, with a 4 to 1 rod, the maximum inertia pressure is, very approximately, fifty times the weight of the reciprocating parts. (C. & G., 1903, H., Sec. B.)

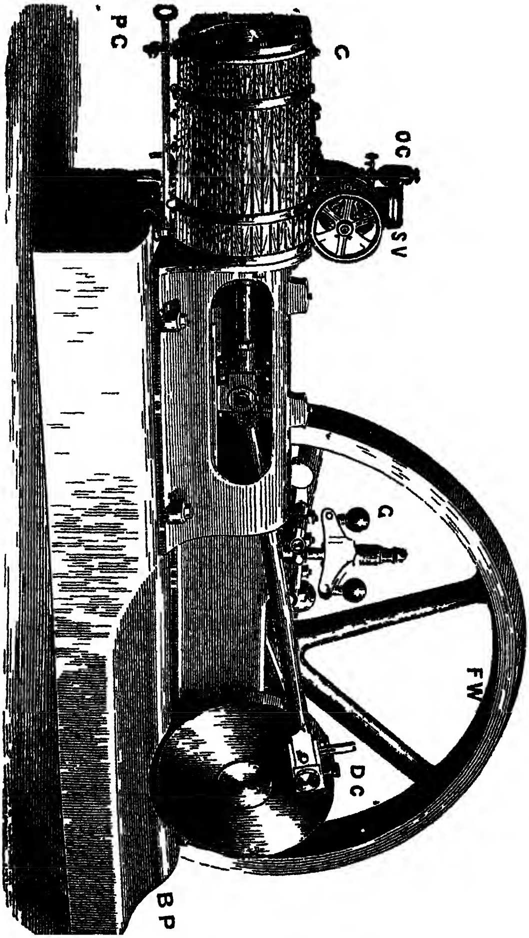
LECTURE XIX.

CONTENTS.—Stationary Land Engines—Horizontal Non-condensing Steam Engine—Horizontal Condensing Steam Engine—Compound Non-condensing Steam Engine with Locomotive Boiler—Coupled Compound Condensing Engine, with Data *re* Crosshead, &c.—Questions.

HAVING discussed in the previous eighteen lectures, the early history of fixed engines up to the beginning of this century, the nature of heat and how it is measured, the generation of steam, as well as its action and distribution in non-condensing, condensing, compound, and multiple expansion engines, we now enter upon the description of a few selected examples of land engines, and of a combined engine with boiler, which have proved to be of good design and workmanship for their purposes.

Within the last eighty years there have been, and there are at present in use, a multitude of styles and types of engines, each more or less specially adapted for different classes of work, such as pumping water from mines, raising water for the supply of towns, draining lands, blowing air into smelting furnaces, driving agricultural machinery, steam cranes, and such like, all of which it is impracticable to treat of fully in this work; for it is impossible in the few remaining lectures at our disposal to do more than indicate the general design with some of the more important details, of the various examples which we have selected. In some instances, we shall give the actual specifications from which the engines were made, as we know from our own experience, that an apprentice or young engineer (unless he is particularly fortunate and happens to be in the drawing office) has little or no chance of perusing and studying specifications, for these things are, as a rule, carefully locked past and treated as private by the heads of firms. We shall also have occasion to devote two lectures to the rise and progress of the Marine Engine, and part of another to that of the Locomotive Engine.

In the present lecture we shall describe three styles of fixed or stationary horizontal land engines, designed by Messrs. Marshall, Sons & Co., Limited, of Gainsborough, which firm has a high reputation for excellence of workmanship and design, brought about by many years of experience and constant attention to special requirements and to small details.



MARSHALL'S HORIZONTAL NON CONDENSING STEAM ENGINE

S V for Stop valve.
C " Cylinder
O C " Oil cup.
P C " Pet cocks

B P for Bed plate
D C " Disk crank
G " Governor.
F W " Fly wheel

Horizontal Non-condensing Engine—The form of engine which is illustrated above, is specially adapted for driving small works, or dynamo machinery, where economy of coal and of water is not of the first or of vital consideration, but where uniform speed, freedom from breakdown, and simplicity of construction are of great consequence It is usually made in sizes varying from 36 to 105 indicated horse-power, and supplied with

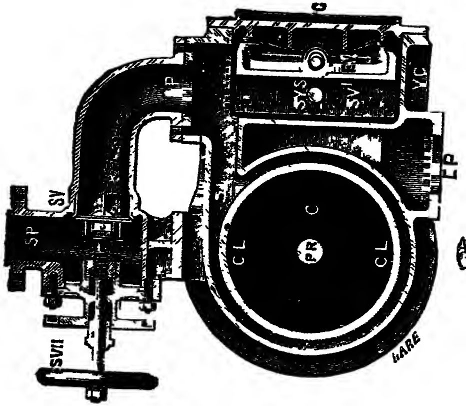
steam from an ordinary Lancashire boiler (see Index), or from a boiler of the multitubular locomotive type (see index), at a pressure of 40 to 80 lbs.,. according to circumstances. As the general construction of this engine is very similar to that of the non-condensing parts in the next set of illustrations, of which we shall give a complete descriptive specification, we need only refer the student to the figure on the last page and the index of parts.

Horizontal Condensing Engine.—This style of engine is much used to drive factories and engine works, where a uniform speed is necessary, and where it is advisable to economise fuel by condensing the steam and returning the feed water warm to the boiler, but where the condensing water is of good quality admitting of the adoption of the jet condenser. It virtually consists of the engine previously illustrated with the addition of a condenser, an air pump, and feed pump. It is also fitted as in the case of the previous engine with Hartnell's Automatic Expansion Gear, which so regulates the cut-off or expansion valve (working on the back of the main slide valve), that steam is admitted to the cylinder in almost direct proportion to the load to be overcome. This ensures an almost perfect uniformity of speed, whether many or few of the factory machines are set to work, or whether few or many of the electric lights are in circuit when these engines are applied to driving dynamos. The construction and action of this engine will be best understood by following the drawings and specification for one of 80 I.H.P.

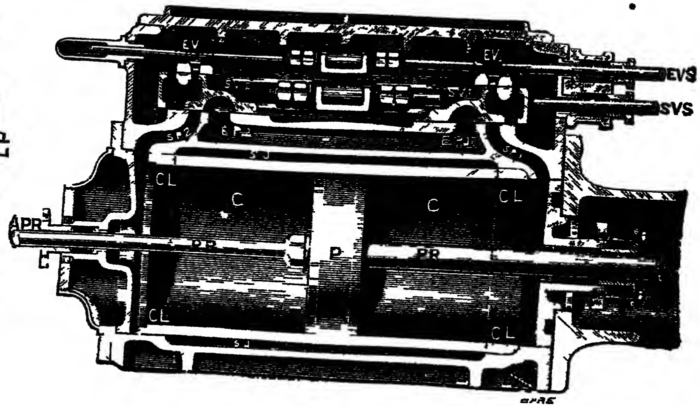
General Construction.—The engine is erected on a heavy bed-plate, B P, of hollow girder pattern, truly planed on the underneath surface. This bed-plate is arranged so as to form at one end the front cover for the cylinder, C, and at the other end the main bearing for the crank shaft, C S. Sliding surfaces for the cross-head, C H, are embodied in the same casting. The crank shaft is constructed with a disk crank, D C, and a pin for the attachment of the connecting-rod, C R. The outer bearing for the crank shaft is on a separate foundation with plummer block, P B. Sufficient room is afforded on the crank shaft by the side of the fly-wheel, F W, for the application of a pulley to give off the whole or a portion of the power if required.

The engine in all its parts is of ample strength for working with steam supplied at 80 lbs. pressure, and of developing 80 indicated horse-power at 70 revolutions per minute, with a cut-off at $\frac{3}{8}$ stroke, and a mean pressure of 58 lbs. in the cylinder. When supplied with dry steam, the average consumption of feed water in the form of steam is 25 lbs. an hour per indicated horse-power.

SCALE 3/4" ONE FOOT



CROSS SECTION

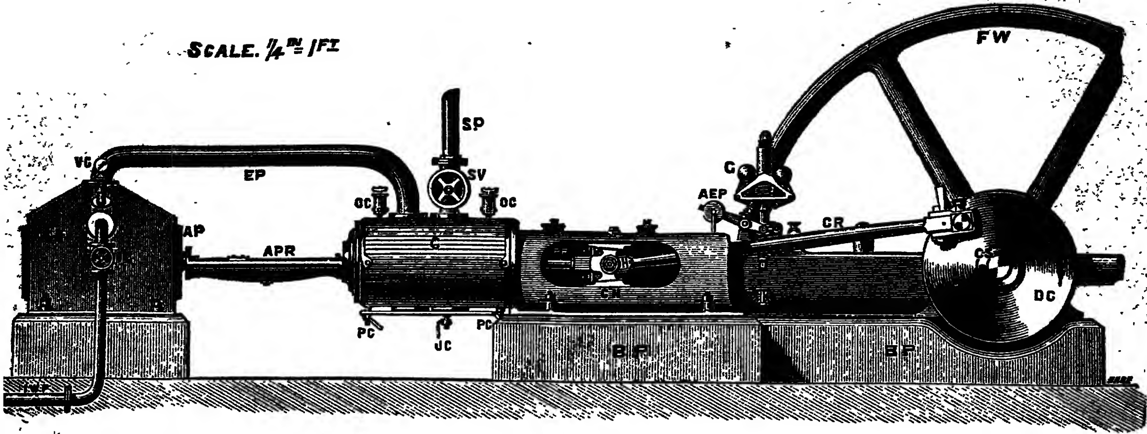


LONGITUDINAL SECTION.

Cylinder and Slide Valves—The cylinder, C, is of cold blast iron, with a steam jacket, S J. The working cylinder barrel or cylinder liner, C L, is of special hardness, cast separately, and securely forced into the main casting of the cylinder. This liner is truly bored out to a diameter of 14½ inches, the stroke of the piston, P, being 30 inches The main slide valves, S V_1, S V_2, as well as the expansion valves, E V_1, E V_2, are all of the same class of iron as the cylinder, in order to insure uniformity of wear. Both the slide valve spindle, S V S, and the expansion valve spindle, E V S, are of steel.

In the cross section, the main steam and exhaust pipes are marked respectively, S P, and E P, while

SCALE. ¼ in = 1 Ft



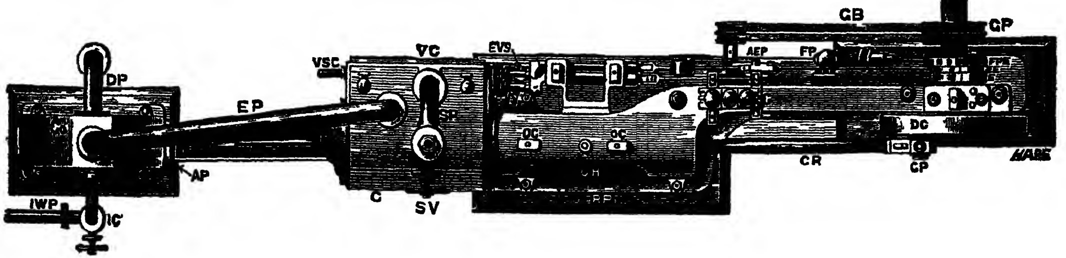
SIDE ELEVATION.—MARSHALL'S HORIZONTAL CONDENSING STEAM ENGINE, WITH AUTOMATIC EXPANSION GEAR.

INDEX TO SIDE ELEVATION AND PLAN.

S P	for	Steam pipe.	C	for	Cylinder.	Co	for	Condenser.
S V	,,	Stop valve.	J C	,,	,, jacket cocks.	I C	,,	Injection cock.
V C	,,	Valve casing.	P C	,,	,, pet cocks.	I W P	,,	,, water pipe.
O C	,,	,, ,, oil cups.	E P	,,	,, exhaust pipe	D P	,,	Discharge pipe
E V S	,,	Expansion valve spindle.	A P	,,	Air pump.	V G	,,	Vacuum gauge.
V S	,,	(Main) valve spindle.	A P R	,,	,, ,, rod.			

INDEX TO SIDE ELEVATION AND PLAN—*Continued.*

B P	for	Bed plate.	E	for	Eccentric for slide valve.
P R	,,	Piston-rod.	E E	,,	Expansion eccentric.
C H	,,	,, ,, Crosshead.	A E P	,,	Automatic expansion gear pulley.
O C	,,	Oil cup for ,,	G	,,	Governor.
C R	,,	Connecting-rod.	G B	,,	Governor belts.
D C	,,	Disk crank.	G P	,,	Governor pulleys.
C S	,,	Crank shaft.	F P E	,,	Feed pump eccentric.
F W	,,	Fly-wheel.	F P	,,	Feed pump.
P B	,,	Plummer block.			



PLAN.—MARSHALL'S HORIZONTAL CONDENSING STEAM ENGINE WITH AUTOMATIC EXPANSION GEAR.

in the longitudinal section, the two steam and two exhaust ports close to the main slide valves, are marked respectively, $S P_1$, $S P_2$, and $E P_1$, $E P_2$.

Automatic lubricators or oil cups, O C (see general elevation and plan), are fixed into the valve casing, V C, so as to thoroughly lubricate the working parts of the slide valves, and the oil being carried forward with the steam, the piston is thereby also lubricated. An efficient drain cock, J C, for draining the steam jacket, and the valve chest as well as drain or pet cocks, P C, for the cylinder barrel are provided. The cylinder is lagged with teak, held in position by brass screws.

Stop-Valve Chest.—The steam stop-valve chest containing the stop valve, S V, which admits steam from the boiler to the slide valve casing, V C, and steam jacket, S J, is shown in section in the cross section of the cylinder. It is bolted to the valve casing in a convenient position for draining the main steam pipe, S P. The stop valve is a wing valve with a suitable seat, both of gun-metal, and is fitted with a brass spindle and screw, kept steam tight by a stuffing-box with brass gland and studs. On the outer end of the spindle is fixed the stop valve handle or wheel, S V H, whereby the attendant can cut off or admit more or less steam at pleasure from the engine.

Piston.—The piston, P, is made of cast-iron, fitted with L, shaped cast-iron rings, and steel internal spring to compensate for wear.

Piston-Rod.—The piston-rod, P R, and air-pump rod, A P R, are of steel, the former being $2\frac{1}{4}$ inches diameter. They are fixed to the piston by a simple cone and nut.

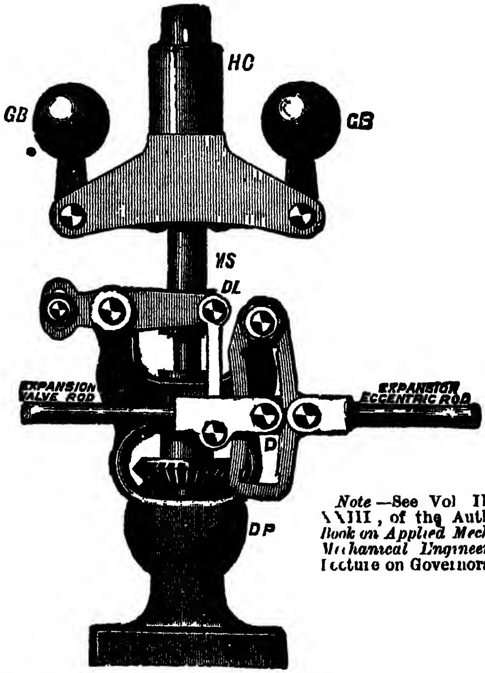
Crosshead.—The crosshead, C H, is of the best malleable iron, finished bright, and has large adjustable bearing surfaces to compensate for wear. It is firmly fixed to the piston-rod, and is provided with a steel gudgeon for attaching it to the connecting-rod. The crosshead guides are of the circular bored type.

Connecting-Rod.—The connecting-rod is of wrought-iron turned and polished. It is fitted with adjustable brasses at both ends as shown.

Crank Shaft.—The crank shaft is of steel, $5\frac{1}{2}$ inches diameter at the bearings. A polished cast-iron disk properly counter-weighted and fitted with a steel crank pin, is forced on to one end of the crank shaft by hydraulic pressure. The crank shaft is supported by two extra long gun-metal bearings, the crank bearing being sustained directly by the engine bed plate, B P, while the outer one is fitted into a plummer block, P B, placed on a separate foundation outside the fly-wheel. Both bearings are made adjustable to follow up the wear.

Fly-wheel.—The fly-wheel is of cast iron, 11 feet in diameter, and 14 inches wide on the face, with arms of strength proportionate to the weight of the rim and the stresses brought to bear on it while working. It is usually turned with the necessary curvature on the periphery so as to receive a driving belt. The boss is bored out and key-wayed to suit the crank shaft. The engine is usually adjusted so that the top part of the fly-wheel revolves from the cylinder unless otherwise specified for.

Governor and Automatic Expansion Gear.—This arrangement consists of (see first the general views in this lecture) the



Note —See Vol II, Lecture XXIII, of the Author's *Text-Book on Applied Mechanics and Mechanical Engineering* for a Lecture on Governors.

HARTNELL'S GOVERNOR, WITH AUTOMATIC EXPANSION GEAR.

governor, G, driven from two governor pulleys, G P, keyed to the crank shaft, with two belts from them, G B, to the two automatic expansion gear pulleys, A E P, which are keyed to the same spindle as the driving pinion, D P (see the accompanying figure). This pinion, D P, gears with another one

keyed to a vertical spindle, on the other end of which is an arrangement for supporting the two governor balls, G B, G B, fixed to bell crank levers, the whole being rotated along with the vertical spindle. The inner ends of the two bell crank levers, bear on a strong spiral spring, contained in or above the upper extension of the metal tube or sleeve, M S. On the lower end of this metal sleeve is fixed a double collar freely engaged by a forked lever, suspended from which is a drag link, D L, whose lower end is attached to the expansion valve rod. The end of the expansion valve rod engages a die-block, D, which may be pulled up or pushed down throughout the length of the curved link, L, to the centre of which is attached the expansion eccentric rod, whose other end is strapped to the expansion eccentric, E E, keyed in position on the crank shaft.

Consequently, whenever the speed of the engine *exceeds* the normal speed for which it has been set to run at, the two governor balls fly outwards by the extra centrifugal or tangential force, compressing the spiral spring lifting the metal sleeve, M S, drag link, D L, and expansion valve rod with the die-block, D, towards the upper end of the curved link, L, thus diminishing the travel of the expansion valve, and cutting off the steam earlier from the cylinder, which reduces the power and speed of the engine again to the normal. When the speed of the engine *falls below* the normal, the reverse action takes place, for then the tension of the spiral spring overcomes the compressive pressure of the bell crank levers, and presses down the metal sleeve, drag link, and expansion valve rod with die-block, towards the lower end of the curved link, thus increasing the travel of the expansion valve, and cutting off the steam later from the cylinder; which increases the power and speed of the engine again to the normal In this way, only sufficient steam is admitted to the cylinder to develop the power required for the load in circuit, and to maintain an approximately uniform speed of from 2 to 5 per cent. above or below the normal speed, under considerable and frequent variations of load; and further, steam is economised by doing so, while the ordinary but less precise acting throttle valve arrangement is dispensed with. All the working parts of this gear are case-hardened and the pins are of steel.

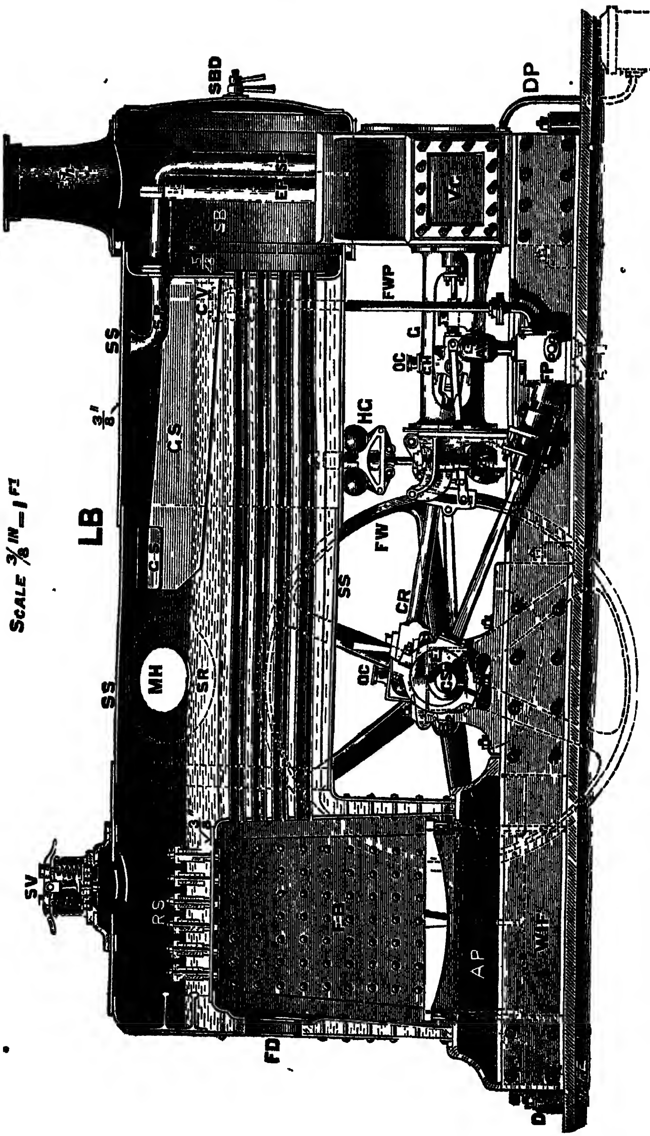
Feed Pump.—The feed force pump, F P, consists of a cast-iron barrel, truly bored out to a diameter to suit a hollow plunger, 3 inches in outside diameter, with a stroke of $3\frac{1}{8}$ inches. It is supplied with the necessary stuffing box, brass-bushed gland and studs, suction and delivery valves with seats, all of gun-

metal (respectively connected by pipes to the condenser hot well and to the boiler as required), and a large cast-iron air vessel. The pump is worked by an eccentric, F P E, keyed to the crank shaft along side of the expansion eccentric.

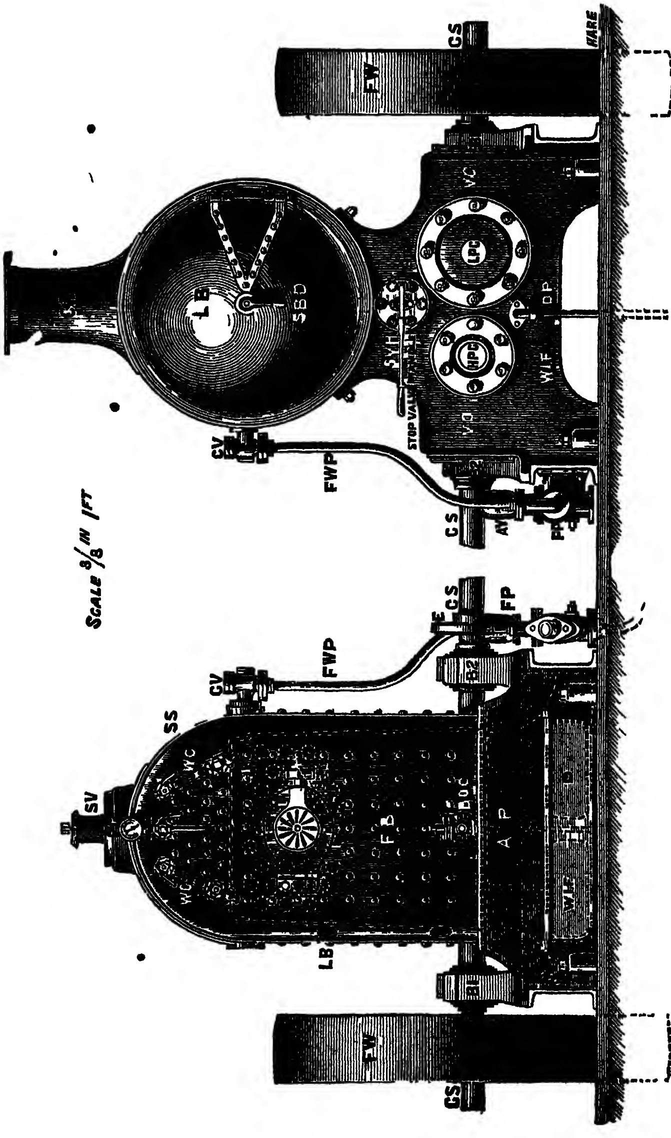
Condenser and Air Pump.—The condenser, Co, consists of a strong cast-iron box of ample size, bolted to and resting upon a cast-iron sole placed behind the steam cylinder. The cast-iron exhaust pipe, E P, joins the exhaust port of the cylinder, and the top of the condenser at its centre. Immediately underneath the latter end of the exhaust pipe, and inside the condenser, is fixed a perforated pipe or rose, leading from the injection cock, I C, and injection water pipe, I W P, which is 2¼ inches internal diameter. An ordinary horizontal double-acting air pump with brass barrel 4⅜ inches internal diameter is fixed in the centre of the condenser. This pump which has the full stroke of engine, is fitted with a brass plunger and air-pump rod, A P R, 1¼ inches diameter, worked direct from the back end extension of the piston-rod as shown. India-rubber suction and delivery valves with brass seating are fixed at each end of the air pump, with a discharge pipe 4½ inches diameter leading from the delivery valves to the hot well. A vacuum gauge, V G, is fitted to the condenser on the same side as the injection cock.

The following table gives the general dimensions, speeds, and horse-powers of such engines:—

Dimensions of Engine.				Revolutions per Minute.	Power				
Cylinder.		Fly Wheel.			Nominal Horse-Power.	Indicated Horse-Power			
						Most economical Load.		Maximum Load.	
Diam.	Stroke.	Diam				Boiler Pressure 60 lbs.	Boiler Pressure 80 lbs.	Boiler Pressure 60 lbs.	Boiler Pressure 80 lbs
In.	In.	Ft	In.						
11	22	7	2	96	12	30	36	42	48
12	24	7	9	88	14	35	42	49	56
13	27	9	0	78	16	40	48	56	64
14½	30	11	0	70	20	50	60	70	80
16	33	12	0	65	25	62	75	87	100
17½	36	13	0	60	30	75	90	105	120
19	36	13	0	60	35	87	105	122	140



SECTIONAL ELEVATION—MARSHALL'S COMPOUND ENGINE AND BOILER.



End Views.—Marshall's Compound Engine and Boiler.

Compound Non-Condensing Engine and Boiler.—This type of combined engine and boiler is very complete and compact. It is, therefore, becoming very popular for driving works and electric light machinery, where want of space or other circumstances prevent the use of a separate engine and boiler. Steam can be raised in this locomotive type of boiler in a very short time, and, owing to the large steam space and heating surface, it keeps steam amply supplied when the engine is working at full power throughout a long and continuous run. The following descriptive specification is for an engine and boiler developing under ordinary circumstances about 50 indicated horse-power, by Messrs. Marshall, Sons & Co., of Gainsborough.

General Construction.—The engine is of an improved construction, mounted on a wrought-iron framing, W I F, underneath a Locomotive Multitubular Boiler, L B, of large capacity, having a steel shell, S S, and a fire-box, F B, of bowling iron. The smoke-box, S B, is bolted to the top flanges of the high- and the low-pressure cylinders, H P C and L P C. The fire-box end rests on a neat ash-pan, A P, fitted with a door, D, for regulating the draught. The cylinders are steam jacketed, and the whole engine is of extra strength throughout to withstand a continuous working steam pressure of 140 lbs. to the square inch, developing 48 indicated horse-power, at 155 revolutions per minute.

Cylinders.—The cylinders are of cold blast iron, with the working barrels of special hardness, cast separately, and tightly forced into the main casting of the steam jacketed cylinders. The cylinders are covered with hair felt cased over with sheet iron. The high-pressure cylinder, H P C, is 8 inches diameter, and the low-pressure cylinder, L P C, is 12¾ inches diameter, each with a stroke of 14 inches. The slide valves are of the same class of iron as the cylinder to insure uniformity of wear. The steam chest and jackets are arranged so as to be effectually drained in a similar manner to that shown and described in the last style of engine, the condensed steam being led away by the drain pipe, D P. Steam is admitted to the valve casing, V C, from the boiler by the steam pipe, S P, on opening the stop valve handle, S V H, and the steam is emitted by the exhaust pipe, E P, up the chimney, Ch.

Pistons, Piston-Rods, and Crossheads are of precisely the same type as described in the last style of engine.

Guides.—The guides, G, are of the circular bored type, bolted to the cylinders at one end, and to the wrought-iron bridge plate at the other end. It is fitted with the necessary oil cup, O C.

Connecting-Rods.—The connecting-rods, C R, are of the best scrap iron, turned and polished, and fitted with large adjustable bearings at each end.

Crank Shaft.—The crank shaft, C S, is made of steel in one piece, without weld, and of sufficient length to take on the fly-wheel, F W, on either end as may be required. It is carried on long gun-metal bearings, B 1, B 2, firmly bolted to the wrought-iron framing, W I F. These bearings are made adjustable horizontally, to follow up the wear.

Fly-wheel.—The fly-wheel, F W, is 5 ft. 6 in. diameter, 9½ in. wide on face. It is constructed exactly in the same way as in the last style of engine.

Governor and Automatic Expansion Gear are applied precisely as in the last style of engine, but to the high-pressure cylinder engine only. It is marked, H G, for Hartnell's Governor.

Feed Pump.—A continuous action force pump, F P, with an air vessel, A V, and worked by an eccentric, E, keyed to the crank shaft is bolted to the side of the engine frame. This pump plunger, suction, delivery valves and taps are all of gun-metal, as well as the check valve, C V, fixed to the side of the boiler, and connected to the pump by the copper feed water pipe, F W P.

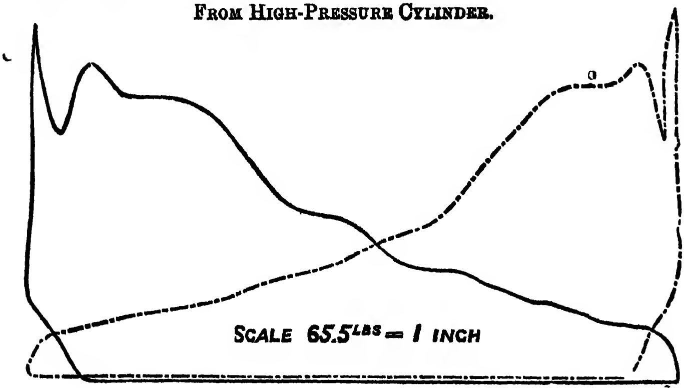
Boiler.—The boiler, L B, is of the locomotive multitubular type, lagged and cased over with sheet iron the whole length. It is of ample capacity for generating and maintaining a continuous supply of steam for the engine when developing full power. The internal fire-box is of suitable dimensions for burning either coal or wood as fuel, and strongly stayed at the ends and sides by screwed stays, and at the top by deep roofing stays, R S. All the boiler plates are planed on their edges, and riveted together by hydraulic machinery. The longitudinal seams are double riveted, and the boiler throughout is of sufficient strength to withstand a continuous working pressure of 140 lbs. to the square inch. Long gusset stays, G S, are riveted between the smoke-box end and the main steel shell, S S. There are 36 high pressure lap-welded iron boiler tubes, 2½ inches external diameter, extending between the fire-box, F B, and the wrought-iron smoke-box, S B. This smoke-box is fitted with a suitable smoke-box door, S B D, furnished with a strap, hinges and fasteners. The firing door, F D, furnace bars, and manhole, M H, are fitted with external strengthening rings, S R. Two spring loaded safety valves, S V, of ample capacity, water-gauges, W G, W G, with gun-metal fittings, two gauge cocks, steam pressure Bourdon gauge, P C, gun-metal blow-off cock, B O C, are provided, as well as a fusible plug in the crown of the fire-box, and a straight chimney, Ch, of wrought-iron 8 feet long.

Indicator Diagrams.—The following set of diagrams were taken from engines made in accordance with the foregoing specification and the working drawings from which the previous figures were reduced, where—

Boiler pressure	= 140 lbs. on square inch.
Diameter of H.P. cyl.	= 8 inches.
Cut-off in ,,	= $\frac{1}{4}$ stroke.
Clearance ,,	= $\frac{1}{12}$ of its volume.
Diameter of L.P. cyl.	= $12\frac{3}{4}$ inches.
Cut-off in ,,	= $\frac{1}{2}$ stroke.
Clearance ,,	= $\frac{1}{13}$ of its volume.
Number of revolutions	= 155 per minute.

The back end is the full line diagram, and the front end the dotted line.

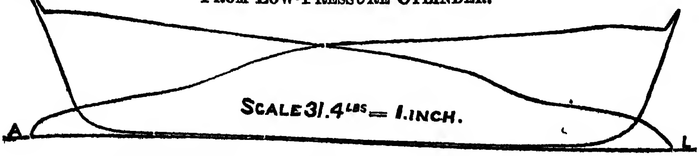
FROM HIGH-PRESSURE CYLINDER.



ATMOSPHERIC LINE.

The irregular line on admission is caused by the indicator not being suited to such a high speed as 155 revolutions per minute.

FROM LOW-PRESSURE CYLINDER.



The following table gives the general dimensions of cylinders, fly-wheels, and the speeds of these engines for different horse-powers:—

Nominal Horse-Power	Cylinders. High Pressure	Cylinders. Low Pressure.	Stroke in Inches.	Revolutions per Minute.	Diameter of Fly-wheel. Ft.	Diameter of Fly-wheel. In.	Indicated Horse-Power given off with Economy.
	Diameter in Inches.	Diameter in Inches.					
8	5½	9	12	180	4	0	26
10	6½	10½	14	155	5	0	33
12	7	11¼	14	155	5	0	40
16	8	12¾	14	155	5	6	52
20	9	14	16	135	6	0	65
25	10	16	18	120	7	0	80
30	11	17½	18	120	7	0	95
40	13	21	24	90	8	0	130

Coupled Compound Horizontal Fixed Condensing Engine, designed and constructed by Messrs. Robey & Co., of Lincoln, and fitted with the Richardson & Rowland Patent Automatic Trip Expansion Gear.

Adapted for, &c.—This type of engine, as illustrated, is specially designed and adapted for driving electric lighting machinery, large factories, mills, &c., where regularity of speed with varying loads, as well as high efficiency in the economy of fuel, is necessary.

Illustrations.—The illustrations are taken from the engine which was employed in the Electric Light Department of the International Exhibition, Glasgow (1888), for driving the dynamos on the north side. It did its work without a single hitch. This engine is now fitted at Messrs. J. & G. Thomson's, of Clydebank, for driving their ship-yard machinery and saw mill, &c.

Fig. 1 (*A*), shows a front elevation, (*B*), plan, and (*C*), end elevation; Fig. 2, longitudinal and cross sections through high-pressure cylinder; Fig. 3, enlarged cross section through high-pressure cylinder at steam admission and exhaust valves; and Fig. 4, an improved form of crosshead and gudgeon pin.

Cylinders and Cut-off.—The cylinders, which are both steam-jacketed, are respectively 18¼″ and 30″ in diameter, with a stroke of 40″. Each cylinder is fitted with the trip valve gear, the cut-off on the high-pressure cylinder being capable of being varied by the governor from *nil* to three-quarters of the stroke, whilst the cut-off on the low-pressure cylinder is variable by hand, and when the engine is running.

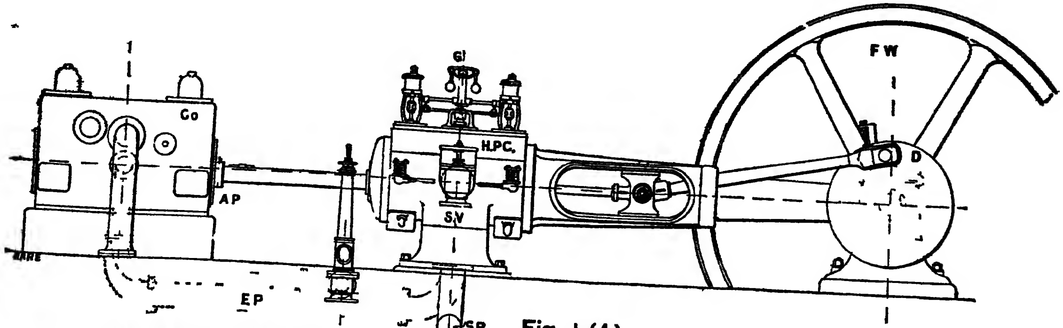


Fig 1 (*A*.)

SIDE ELEVATION OF ROBEY & Co.'s COUPLED COMPOUND HORIZONTAL CONDENSING ENGINES.

Revolutions.—The engines are speeded to give 63 revolutions per minute at an initial pressure of 100 lbs. steam per square inch, and transmit their power (400 I.H.P.) from a flywheel, **F.W.**, 13′ diameter, 24″ wide, and seven tons in weight.

Steam and Exhaust Valves.—Both high- and low-pressure cylinders, **H.P.C.** and **L.P.C.**, have independent admission valves, **A.V.**, arranged on the top, and exhaust valves, **E.V.**, fitted to the bottom of the cylinders (see Figs. 2 and 3). The former consists of double-beat Cornish equilibrium valves fitted to each end of the cylinders, so as to get the shortest possible steam passage, thus enabling the engine to work at all times with an initial pressure as nearly approaching that of the boiler as possible.

The Admission Valves on the high-pressure cylinder are under the direct influence of the governor.

The Exhaust Valves, **E.V.** (Fig. 3), consist of a special arrangement of Corliss slide valve, which gives a quick opening to the exhaust with a very small travel. They are placed underneath the cylinder,

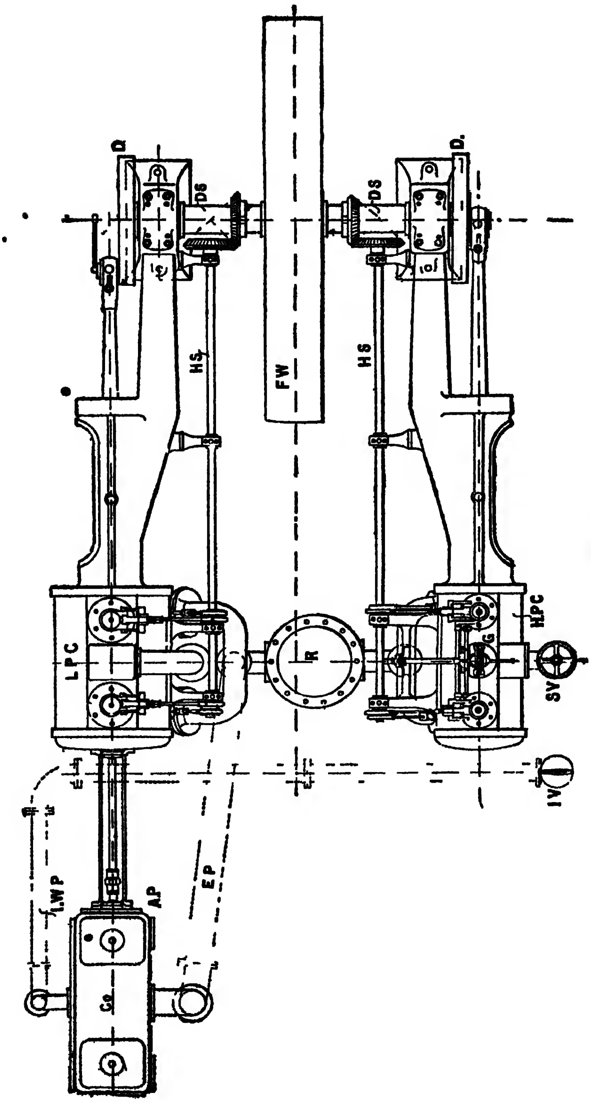


Fig 1 (B.)

Plan of Robey & Co.'s Coupled Compound Horizontal Condensing Engines.

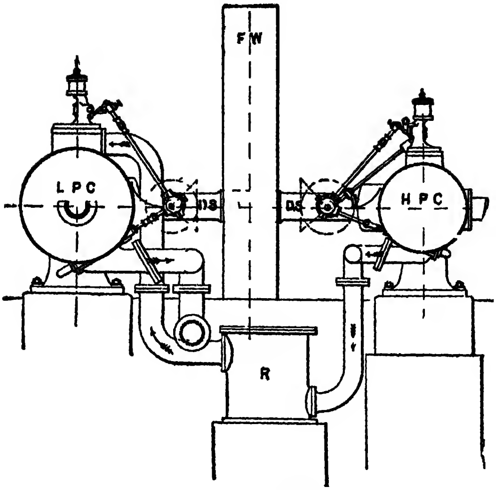


Fig. 1 (*C.*)

in order to efficiently drain the interior, and enable the pistons to work safely with the least possible amount of clearance. They are worked by exhaust eccentrics, E.E., upon the horizontal shaft, H.S., driving the admission valve gear.

Action of Admission Valves and Governor.—Following the action of the steam inlet valves from Figs 2 and 3, it will be noticed that the admission valves, A.V., are lifted and released by trip levers, T.L., actuated by the admission eccentrics, A.E., driven by the horizontal shaft, H.S., rotating at the same speed as the disc shaft, D.S., and running parallel with the engine-bed. The length of time the trip levers are in contact and consequent duration of the admission of steam into the cylinder is regulated by the governor, G, thus automatically varying the grade of expansion to the work being done. The upper portion of the valve spindle, V.S., is attached to an air buffer, A.B., which, assisted by a spiral spring, suddenly closes the valves when relieved from the trip lever.

A very precise action of the valve is obtained by this arrangement, and a very sharp cut-off is consequently insured. To prevent the admission valves, A.V., being forced down too suddenly upon their seats, S, the usual air cushion is formed and regulated by valves in the air buffer, A.B., which are so constructed that, while the admission valves close steam-tight, they yet come upon their seats with checked velocity. The

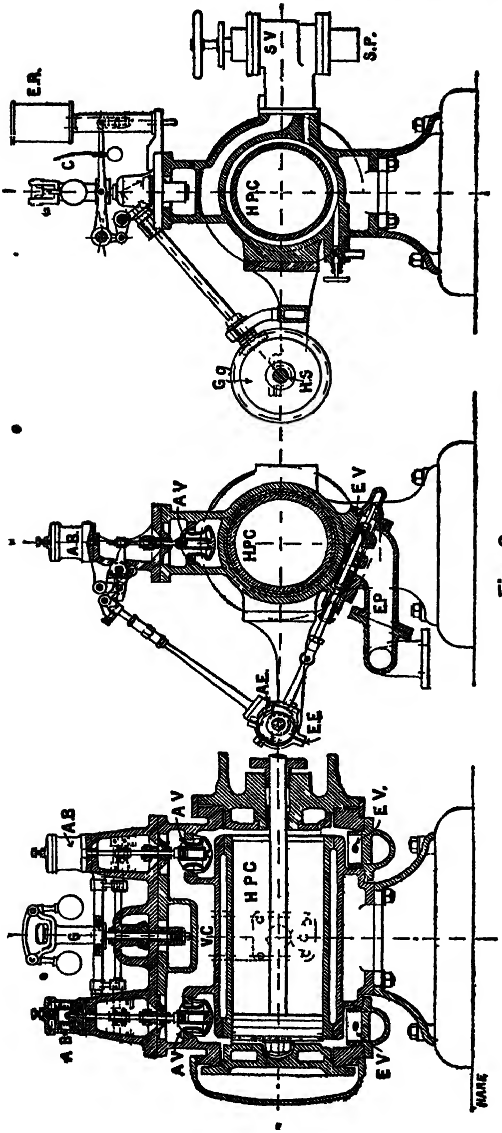


Fig. 2.

Sections through the High-Pressure Cylinder of Robey & Co.'s Coupled Compound Horizontal Condensing Engines.

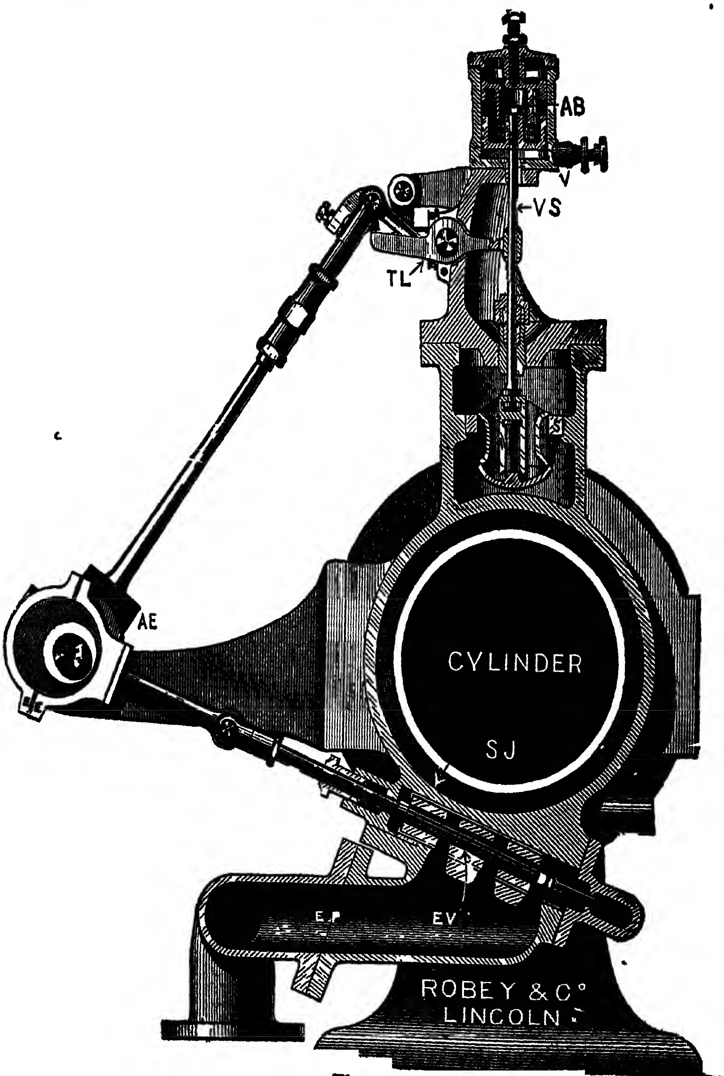


Fig. 8.

RICHARDSON & ROWLAND AUTOMATIC TRIP EXPANSION GEAR.

governor G (Fig. 2), regulating the admission valves, is one of Richardson's patent spring governors, which, being relieved of all working stress, is so constructed as to give a wide range of cut-off with very slight variations in speed. It is driven by gearing, Gg., from the horizontal shaft, H.S. The admission eccentrics, A.E., are so fixed upon the same horizontal shaft, H.S., as to give a constant lead. When used for electric lighting the governor is supplemented by a Richardson-Nevile Patent Electric Regulator, E.R., Fig. 2, which enables the engine to be controlled by the electric current itself, so as to maintain either a constant current or a constant E.M.F. with varying loads.

The valve gear is also arranged so that the engine can be stopped by merely pulling a cord, C, carried to any part of the mill or factory, a provision which is invaluable in case of accident to life or machinery.

Framing.—The engine-frames or bed-plates are of the most solid and substantial character, efficiently resisting the direct thrust and working of the engine, thus securing complete rigidity between the cylinder and main bearings, and efficiently taking up any stresses in the crosshead guides; this design being altogether a great improvement upon the original type of girder engine as first introduced into this country. The bearings, which are extra large, are made in three adjustable parts of Babbit's metal, fitted with suitable lubricators for continuous running.

The steam, in passing from the high-pressure cylinder, H.P.C., to the low-pressure cylinder, L.P.C. (Fig. 1), enters a receiver, R, which is superheated by a current of high-pressure steam from the boiler circulating through a coil of piping placed inside it, thus raising the temperature of the steam previous to its admission into the low-pressure cylinder. The receiver is, in addition, lagged with wood and sheet-iron. The other details need no explanation, as they are similar to those of engines previously explained.

Crosshead: How Made and Fitted.—The crosshead illustrated by Fig. 4 possesses several important features which are worthy of notice. It is made of malleable iron or of cast-steel, and is therefore free from the risk of breaking. The curved surfaces, C.S., which bear on the guides are of hard cast-iron, as this forms the best material for wear. These consist of two plates with projecting pins, shown in dotted lines at P, and are further secured by the screws, S, S, S, S. After these curved surfaces are secured into their places, the whole is turned up true from a mandril fitting into the taper which receives the piston-rod. These bearing plates are designedly left without any means of adjustment by the engine-driver, experience having shown that

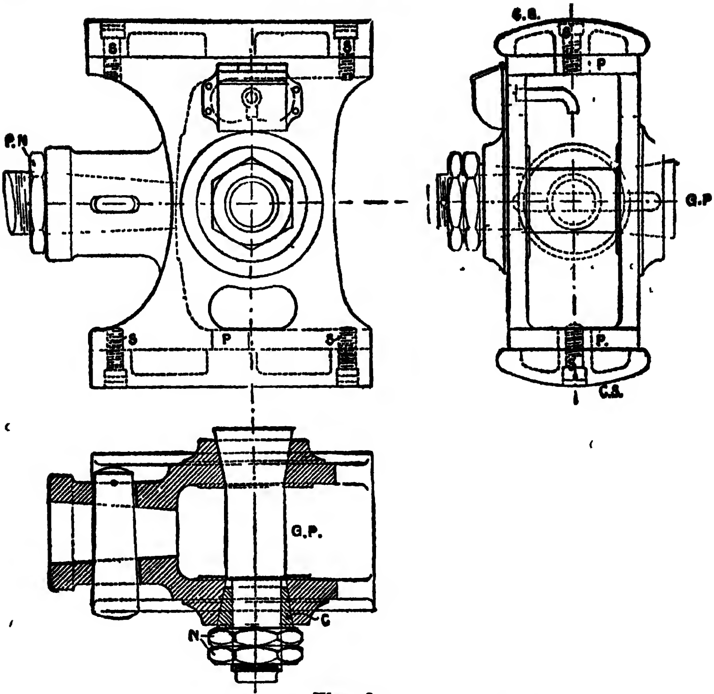


Fig. 4.

ROBEY & CO.'S IMPROVED CROSSHEAD AND GUDGEON PIN.

when such adjustments exist it is more easy to put a crosshead wrong than right. Many crosshead guides have been ruined by screws or wedges being improperly tightened by a careless driver. Should the crosshead shown by Fig. 4 ever get slack, the rubbing surfaces can be packed out by strips of metal, and the exterior again fitted into its place with very little trouble. The surfaces are, however, made so large that there is practically no wear; for guides of this proportion have been known to be in perfect working order at the end of twenty-five years' work. The gudgeon pin, G.P., is a part that has often given trouble, for these pins have to be made to fit so that they can be taken out when required. They are therefore liable to get easily loose. Many methods have been employed to prevent this, but Messrs. Robey & Co. find that shown by Fig. 4 to be the best. The crosshead is bored out taper on its two cheeks, the tapers being in opposite directions. Into one of these the tapered head of the

steel pin, G.P., fits, the other end being turned parallel and is surrounded by a taper steel cotter, C. The cotter is split longitudinally, and is forced by the nuts, N. N., tightly into the coned hole, and at the same time is equally forced to fit tightly upon the pin. It is thus so firmly fixed that it is practically solid with the crosshead when the nuts are screwed up, whilst, when required to be removed, it comes out with the greatest ease. The piston-rod is secured by a cotter into the taper neck of the crosshead in the usual way. For the purpose of removing it (when the cotter is driven out) the piston-rod is provided with a fine thread and a hardened nut, P.N., just behind the crosshead. When this nut is screwed up to the crosshead the rod is drawn without any difficulty. Without such a provision as this, much loss of time and temper is often occasioned.

LECTURE XIX.—QUESTIONS.

1. Explain in general terms the difference between (1) a simple non-condensing engine; (2) a condensing engine; (3) a compound non-condensing engine.

2. Give free-hand sketches (outside elevation and plan) of a horizontal condensing engine, with a complete index of parts, and the uses as well as materials of which each part is composed.

3. Describe with a sketch the construction of a piston, piston-rod, crosshead, and connecting-rod for a horizontal land engine, and show how the several parts are fitted together, and of what materials each part is composed, and why.

4. Sketch a longitudinal section and cross-section through the cylinder of a horizontal condensing engine with expansion valve. Give a complete index of the various parts with the materials of which they are composed. Show how the steam passes into and out of the cylinder, and explain how the piston, piston-rod, and valve spindles are kept steam tight.

5. Describe with sketches and index of parts a compound non-condensing stationary land engine, as usually fitted underneath a locomotive multitubular boiler.

6. What is meant by "Automatic Expansion Gear?" Give the necessary sketches with index of parts and concise explanation to enable a person to understand its complete action, and point out the advantages usually claimed for it over an ordinary governor and throttle valve.

7. Construct scales to suit the indicator diagrams given at p. 328 of this lecture, and divide the diagrams, as well as plot them down to one scale by the method explained and illustrated in the case of H.M.S. *Boadicea*, with the three steam expansion curves. Find also, the mean horse-power developed by each cylinder, and the weight of steam used by the engine per horse power hour on the assumption that the steam is "dry saturated steam."

8. Describe a horizontal factory engine which is to work expansively and with condensation. Enumerate the principal parts, and make the sketches necessary for showing the internal construction

9 Sketch a section through a compound cylinder horizontal factory engine. Show the valves for the distribution of steam, and explain generally the advantages of this form of construction.

LECTURE XX.*

CONTENTS.—Short History and General Description of the Corliss Valve Engine—Special Features of the Corliss Cylinders and Positions of the Valves—Different Types of Corliss Valve Gears—Shape and Construction of Steam and Exhaust Valves—The Original Form of Corliss Trip Gear—Simultaneous and Relative Movements of the Wrist-Plate and Valve Levers—General Description of the Connections between and Movements of Eccentric, Wrist-Plate, Valves and Governor—Farcot-Corliss Valve Gear—Reynolds-Corliss Valve Gear—Double Eccentric Gears—The Seigrist System of Automatic Lubrication for Large Engines—Manipulation of the Oil—Cylinder Lubricator—Compound Engine with Automatic Lubrication—Triple-expansion Engine with Automatic Lubrication—Results with Superheated Steam—Necessary Precautions to be observed with Superheaters and with Highly Superheated Steam—Tests of Willans Engine with Ordinary Steam—Percentage Gain in Steam and in B.T.U. when supplied with Superheated Steam—The Willans Central Valve Engine—Criticism of the Farcot Corliss Cylinder and Position of Valves—Questions

Short History and General Description of the Corliss Valve Engine.†—In the year 1849 an American engineer, Mr. G. H. Corliss, patented and constructed this type of engine, which still bears his name. In 1859 the first engine imported into this country from America was set to work at the Stoneywood Paper Works, near Aberdeen,‡ and the first licencee for the manufacture of Corliss engines in Great Britain was Mr. Robert Douglas, of Kirkcaldy. The firm of Douglas & Grant have, since 1863,

* See Lecture XX., continued in Appendix E, with full description of Dobson's Trip Gear and Cole, Marchent & Morley's Compound Superheated Steam Engine.

† No mention was made of Corliss engines or valves in either the first edition of Prof. Rankine's *Manual on the Steam Engine*, published by Charles Griffin & Co. in 1859, or in John Bourne's *Treatise* and his *Catechism of the Steam Engine* up to 1865. The first published explanation in this country appears in the *Transactions of the Institution of Engineers and Shipbuilders in Scotland*, vol. vii., 1863 4, by W. Inglis, and again, by the same engineer, in the *Proc. Inst. M.E.* for 1868, in a paper on "The Corliss Expansion Valve Gear for Stationary Engines." Also, see *The Steam Engine*, by D. K. Clark, vol. iii., Blackie & Son, 1890; *Valves and Valve-Gearing*, by Charles Hurst, 1902, Charles Griffin & Co.

‡ Whilst writing the above (from the mere recollection of having been taken as a boy to see this engine in 1860), I have received a letter, dated March 10th, 1904, from Mr. A. G. Groundwater, the chief engineer of these works, in which he says—"I have much pleasure in informing you that the late Mr. A. G. Pirie (head partner) brought the first Corliss steam engine from America to this country and started it here in 1859. It was called the 'Yankee,' and it worked continuously for 32 years, driving part of the works until we got larger engines, when it was not thrown away, but connected up for driving our electrical installation, and is now running as sweetly as ever." Forty four years at work in a place where everything is of the best is a very good testimonial.

made many horizontal mill engines with the latest up-to-date improvements in Corliss valve gear for all parts of the world.

The distinctive feature of this engine lies solely in the distribution and the special method of working its valves. Many of the most eminent British engineers showed a decided reluctance in believing that any such arrangement could surpass the reciprocating flat slide-valve. But nowadays, some of the finest and largest mill engines, where a steady-running, economical steam prime mover is desired, as well as a number of the latest, most powerful and best "Central Station Engines for Electric Light and Tramway Power Installations" are constructed with one or other of the many patented Corliss modifications.

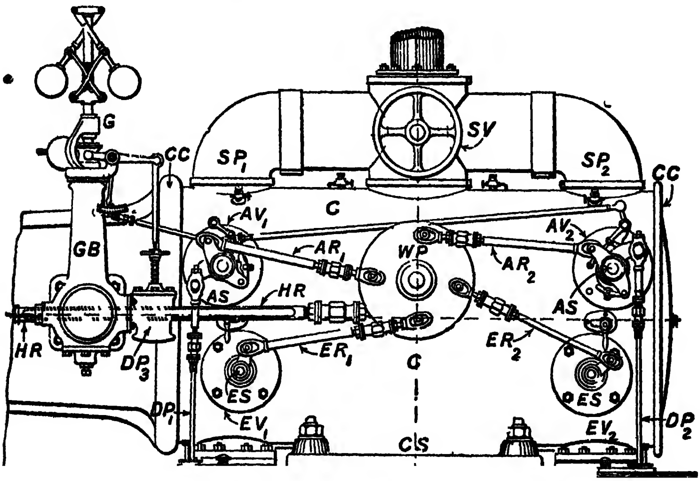


FIG. 1 —OUTSIDE VIEW OF FARCOT-CORLISS VALVE GEAR.

As will be seen from Figs. 1 to 10, with index to parts, the Corliss engine still retains the original and distinctive feature of having four, independent, segment-shaped, arc-faced valves, placed at or near the ends of the steam cylinder, with the peculiar method of rocking these over bored-out steam ports, and of automatically regulating the speed due to altering the point of cut-off by means of the governor.

When the stop valve, S V (Figs. 1, 2), is opened, steam from the boiler fills the steam pipes, SP_1, SP_2, up to the admission valves, AV_1 and AV_2 When either of these steam valves uncovers its port, steam enters one end of the cylinder, C, and

forces forward the piston, P. During the return stroke of the piston, this steam leaves the bottom of the same cylinder end by the exhaust valve, EV_1 or EV_2, and its exhaust pipe, EP_1 or EP_2, direct for the condenser if the engine be a simple condensing one, or for the receivor of the next cylinder should it be of the compound type.*

Special Features of Corliss Cylinders and Positions of the Valves.†—From an inspection of Fig. 2, it will be seen :—

1. That the steam admission valves, AV_1 and AV_2, are situated in the cylinder covers, surrounded by the live, fresh steam, whereby they are kept as hot as possible.

2. That a minimum distance exists between the curved working surfaces of the valve faces, AV_1, AV_2, and the inside of

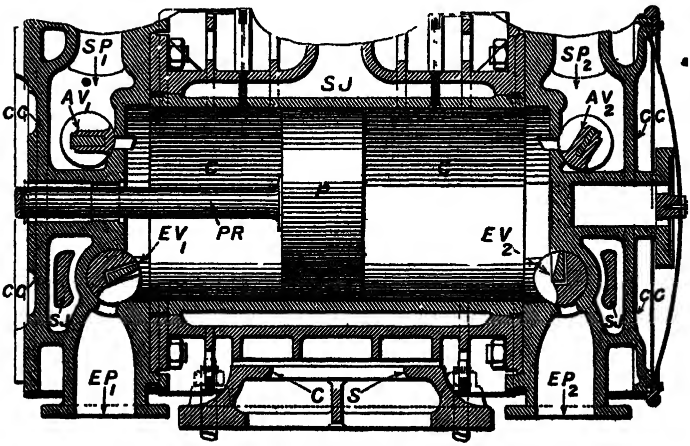


FIG. 2.—LONGITUDINAL VERTICAL SECTION OF CYLINDER IN FIG. 1.

INDEX TO PARTS FOR FIGS. 1 AND 2.

$SP_{1,2}$	for Steam pipes.	CC	for Cylinder covers.
SV	,, Stop valve.	$EV_{1,2}$	,, Exhaust valves.
$AV_{1,2}$	,, Admission valves.	$EP_{1,2}$	,, Exhaust pipes.
C	,, Cylinder.	CS	,, Central sole-plate.
P	, Piston.	SJ	,, Steam jackets.
PR	,, Piston-rod.		

* The exhaust valves, EV_1 and EV_2, should have been placed so as to clear the piston at the end of its strokes. But this drawing was made direct from the original, in which the designer has forgotten, that if one or other of the motion-rods for these valves broke it would be awkward for the valves.

† *See Index or end of this lecture for further criticism.*—A. J.

the cylinder, C, as represented by the inner thickness of the cylinder covers. Further, that the faces of these valves are kept up to their working surfaces by the pressure of the steam upon the backs of the former, and the way in which they are connected by a rectangular fitted slot to their admission valve spindles, A S (Fig. 1)

3. That to all intents and purposes, the clearance spaces between the ends of the piston and the inner faces of the cylinder cover can be reduced to a minimum, by good design, workmanship and the necessary cushioning of the exhaust steam.

4 That the exhaust valves, E V_1 and E V_2, are also placed in the cylinder covers, but as far away from the steam valves as possible. The well-known pernicious cooling effects and wasteful accompanying initial condensation, which is met with in ordinary slide valves, due to the colder exhaust passing out by the same port and through the same valve as it entered by, is thus neatly and effectually avoided.

5. In the case of horizontal cylinders, any condensation which may take place therein, can readily drain down to the exhaust ports and be entirely swept out through E P_1 or E P_2 during each exhaust stroke, from the fact, that the exhaust valves are situated at the lower side and extreme ends of these cylinders.

6. That the complete cylinder barrel and its ends are surrounded with steam jacket spaces, S J, which may be kept always full of live, fresh, hot steam from the boiler, whilst any condensation which takes place in these jackets may be easily drained off into the condenser hot well by special cocks and pipes.

7. That the cylinder as a whole is fixed to a central sole-plate, C S, so that it can expand or contract more or less freely to or from either end without undergoing very severe stresses due to great changes of temperature. Such an arrangement should permit of the free use of superheated steam, if the valves did not "warp" and the working surfaces could be made to withstand its action. (See Index for Dobson's Vertical Trip Valves.)

It will thus be seen, that the cylinder and valves of the Corliss engine have been designed upon sound scientific principles, with a view to steam economy. We shall now consider how far the action of the valve-motion gear and the governor support this design in the same direction.

Different Types of Corliss Valve Gears.—These may be classified under three main types:—

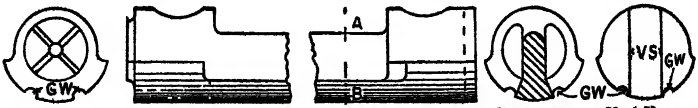
1. *Valve Gears without any "trip"* or disengaging mechanism for regulating the point of cut-off by means of a governor. In

this case, all four valves have a positive connection with the eccentric, their travel is constant and the "point of cut-off" is invariable. Any governing for speed and load is done by a governor acting upon an ordinary throttle valve. This kind was never much used.

2. *Single Eccentric Gears* with "trip" motion for the front and back steam valves. In this case, the two steam valves are so rocked by the eccentric's motion, through its connection with the wrist-plate, &c., against the resistance of springs as to uncover their steam port openings as quickly as possible. Their connection with the eccentric is then released by the governor, which automatically determines the "point of cut-off" according to the speed and load. The two exhaust valves are worked in the same way as case 1 (see Figs. 1 to 4), and are always rocked full open whatever may be the point of cut-off in the steam valves or the load on the engine.

3. *Double Eccentric Gears* with "trip" motion. One eccentric works the two steam valves in the same manner as in case 2, whilst another eccentric is devoted to working the two exhaust valves. Here the periods of steam, cut-off, exhaust, and cushioning may be adjusted before starting the engine by setting the valve spindles, driving levers and connecting-rods to the wrist-plate, &c. This type is now the kind most frequently made in this country for both large horizontal and vertical engines, as shown by the illustration (Fig. 9) of the Reynolds-Corliss valve gear on the high-pressure cylinder for one of the 5,000 I.H.P. engines of the Glasgow Tramways at the Pinkston Central Power Station.

Shape and Construction of Steam and Exhaust Valves.*—As will be seen from an inspection of Figs. 2, 3 and 4, both the



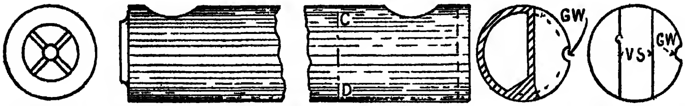
End View. *Outside Longitudinal View.* *Cross section through A B.* *End View.*

FIG. 3.—CORLISS STEAM VALVE.

steam and the exhaust valves are made of cast iron, turned and shaped as indicated by the end views and cross-sections. It will

* As a rule, illustrations of Corliss valves and their gears omit to show longitudinal and more than one cross-section of the steam and exhaust valves, as well as clear indications of how the trip action works. Students will, however, find detailed drawings in the *Machine Construction Book*, by Cryer and Jordan, and in *Engineering* for September 13th, 1895, and October 31st, 1902.

be observed, that examples 3 and 4 are made as light as possible consistent with the necessary strength, and that they differ from Fig. 2 in the method of connection to their steel valve spindles. Figs. 3 and 4 show slots along their right-hand ends at V S, to



End View. *Outside Longitudinal View.* *Cross section through C D.* *End View.*

FIG. 4.—CORLISS EXHAUST VALVE.

receive the extended tongue of the valve spindle in the same way that the head of a screw bolt is slotted to receive the point of the screwdriver. They also have half-circle gutter ways, G W, which form self lubricating channels to carry any oil or condensed steam for the turned end-bearing surfaces.

The Original Form of Corliss Trip Gear.—As the present-day forms of trip gear are somewhat complicated and difficult to understand from a mere inspection and description of drawings, we have selected, as a preliminary diagrammatic view, the original form as devised and applied by Mr. Corliss, and as described by Mr. William Inglis in his paper to The Institution of Mechanical Engineers.

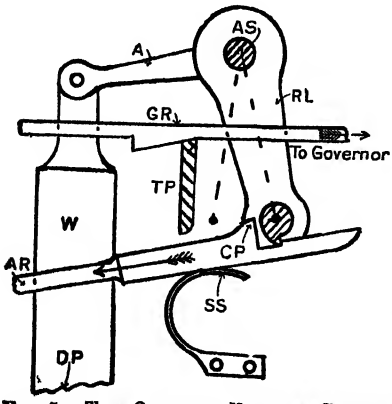


FIG. 5.—THE ORIGINAL FORM OF CORLISS TRIP GEAR.

Here, a weight, W, is attached to the lever, A, on the admission valve spindle, A S, whose rocking lever, R L, is shown in gear with the admission valve rod, A R, from the rocking wrist-plate shown in Figs. 1, 6 and 7. The fixed curved steel spring, S S, keeps the upper end of A R, with its catch part, C P, hard against the catch pin at the outer end of R L, until the upper back curve on C P comes into contact with the trip plate, T P. Should the speed of the engine increase above the normal, then the governor balls are moved outwards by centrifugal force,

thus pulling the governor rod, G R, forward in the direction shown by the arrow (→ to governor). The forcing down of the trip plate, T P, by this inclined wedge-piece on the under side of G R causes the trip plate, T P, to come into earlier contact with the back of C P, and thus releases A R from the eccentric drive sooner in the piston's stroke. This permits the weight, W, to fall quickly at first and to close the steam valve sharply, but its further movement is cushioned or slowed down by means of a dash pot, D P, to which the lower end of W is connected.

In this simple but effective manner, the governor automatically limits the speed of the engine between certain extremes, by tripping or releasing or disengaging the wrist-plate motion rod, and thus permits the steam valve to cut off steam early, if the speed tends to increase, or late, if the speed has been reduced either by an increased load or diminished steam pressure.

Simultaneous and Relative Movements of the Wrist-Plate and Valve Levers.—In Fig. 6, we have shown an educational

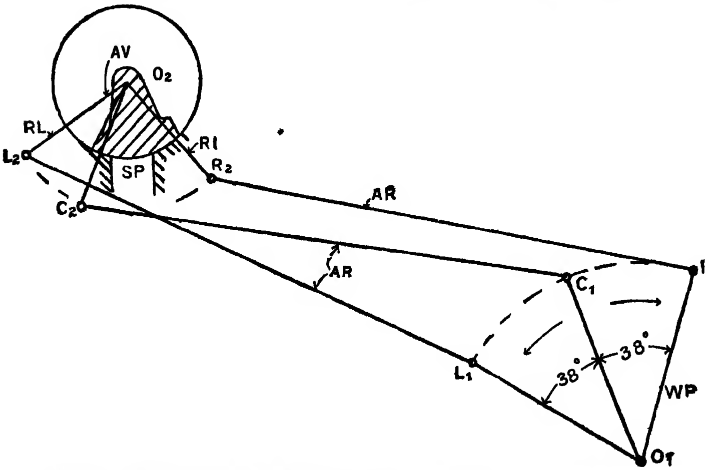
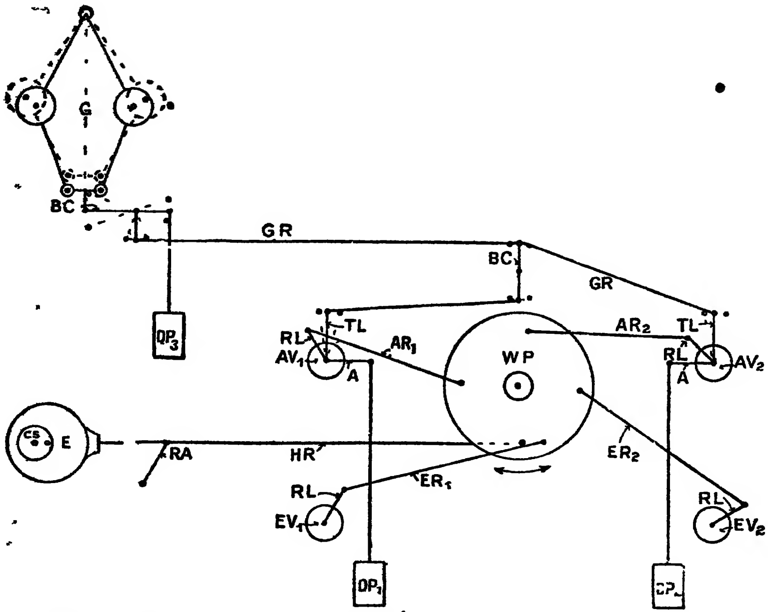


FIG. 6.—MOTIONS OF THE WRIST-PLATE AND CORLISS STEAM VALVE.

diagram, but not to scale, of the angles through which a wrist-plate, W P, and a rocking lever, R L, of an admission valve, AV, move simultaneously, due to their being rigidly connected by an admission rod, A R, in order to illustrate how the wrist-plate modifies the motion of the steam valve, from what it would otherwise be, if the admission lever of the latter was connected direct to the eccentric rod.

Let $O_1 C_1$ represent the centre line of the wrist-plate, W P, when it is in the middle of its rocking motion, as communicated to it by its connection with the engine eccentric; then, the centre line of the valve's rocking lever, R L, will lie along the line, $O_2 C_2$, due to the rigid connection, $C_1 C_2$, of the admission rod, A R. Now, let the wrist-plate be turned to the *right* by the eccentric's motion through the angle, $C_1 O_1 R_1$, of say 38°, then the valve's rocking lever, R L, will have moved during the same time through the *larger* angle, $C_2 O_2 R_2$, of say 63°. This shows that the valve face was moved quicker through its lap + lead + opening of steam port, SP, than it would have been moved, had it been connected directly to the eccentric. Again, the same quicker motion would have taken place whilst shutting the steam port, S P, through the same angle. But, as the wrist-plate moves to the left of the centre line, $O_1 C_1$, of its motion through the same angle of 38° into the position $O_1 L_1$, the valve's rocking lever, R L, only moves in the same time through an angle of 33° into the position $O_2 L_2$. It is thus apparent, that the mere introduction of a wrist plate and the judicious selection of a good location for the admission-rod pin at C_1 on the wrist-plate, W P, will cause a quick opening of the valve for the admission of steam to the cylinder, whilst the movement of the said valve will be slow when "dwelling" over the rest of its ineffective movement. Of course, the introduction of the trip motion shuts the steam valve still more quickly than if it retained its rigid connection with the wrist-plate throughout its whole to-and-fro travel. The Americans thought, at first, that it was this very quick cut-off due to the trip gear, which caused the extra economy in steam; but now, that idea is exploded, since it is the total area of an indicator diagram which is a measure of the work done for a certain weight of steam supplied per unit of time.

General Description of the Connections Between and Movements of Eccentric, Wrist-Plate, Valves and Governor.—From what has been said, and by a comparison of Figs 1 and 7 with the index attached to the latter, the student will at once understand from the centre-line connections between these several parts how the movements are imparted by the eccentric, E, to the wrist-plate, W P, through the joint at the upper end of the radius arm, R A, and hook rod, H R. Also, how the two short connecting-rods, A R_1 and A R_2, transmit motion from the wrist-plate, W P, to the rocking levers, R L, of the admission steam valves, A V_1, A V_2. And, in the same way, how the rods, E R_1 and E R_2, do the same for the exhaust valves, E V_1 and E V_2. Further, it will be seen how the governor, G, is connected by



INDEX COMMON TO FIGS. 7 AND 8.

CS	for	Crank shaft.
E	,,	Eccentric.
RA	,,	Radius arm.
HR	,,	Hook rod.
WP	,,	Wrist-plate.
$AR_{1,2}$	,,	Admission valve rods.
$ER_{1,2}$	,,	Exhaust valve rods.
RL	,,	Rocking levers.
$AV_{1,2}$	,,	Admission valves.
$EV_{1,2}$	,,	Exhaust valves.
G	,,	Governor.
BC	,,	Bell cranks.
GR	,,	Governor rod.
TL	,,	Trip levers.
A	,,	Arms.
$DP_{1,2,3}$	,,	Dash pots.

FIG. 7.—DIAGRAMMATIC CENTRE-LINE VIEW OF CONNECTIONS AND MOVEMENTS OF THE CORLISS VALVE GEAR.

bell cranks, B C, and governor rods, G R, and trip levers, T L, to their respective dash pots, $D P_1$, $D P_2$, by their arms, A, and to its own dash pot, $D P_3$.

It will be evident, how the steam valves $A V_1$ and $A V_2$, replace the front and back working edges of the ordinary reciprocating slide valve as far as lead, or admission of steam and cut-off to their respective steam ports are concerned. Also, how the exhaust valves $E V_1$ and $E V_2$ take the place of the exhaust edges in the ordinary slide valve in determining the points of release, exhaust, and cushioning of the steam in a cylinder.

Farcot-Corliss Valve Gear.—Having mastered the general action of the several elements which come into play in regulating the admission and exhaust steam, the student will now be prepared to tackle the details of this example of levers and cams which serve to actuate the admission valves by special reference to Figs. 1 and 8. The latter figure shows these details by four views, which represent the gear for the right-hand admission valve, AV_2. Here, the rocking lever, $R L_2$, is mounted loose on the projecting boss of the arm, A The lower end of $R L_2$ carries a trip or catch plate, C P, which is constantly drawn towards the admission valve spindle, A S, by a spring. The precise position of this spring is shown in the hole opposite to the letters C P on the lower left hand figure or sectional plan through F_2 to centre of A S and then to O. Upon A S is keyed the arm, A, the outer right-hand end of which is connected by the vertical rod to a spring contained in its dash pot, $D P_2$. It is this spring which closes the admission valve quickly over its steam port, and the dash pot itself cushions or arrests the ending of the downward motion. The left-hand end of the arm, A, carries a lower catch plate, C P, which engages with the upper catch plate, C P, connected to lower end of $R L_2$. Both catch plates are clearly seen and marked C P on the upper left-hand end view of Fig. 8.

It will now be readily understood how the admission spindle, A S, is suddenly turned and admission valve, $A V_2$, closed, whenever these two hardened steel catch plates are disengaged from contact with each other. This disengagement is effected by the governor-rod acting through the bell crank, B C, on the two cam links, C L, which are fixed to loose cams, K_1 and K_2, as shown on the right-hand lower section through x.y. Now, looking at the lower left-hand section, it will be seen, that when the projection of cam, K_2, comes into contact with the finger F_2, it presses the same outwards from the centre of A S until the upper catch plate, C P, is freed or disengaged from the lower one. This allows

the spring and dash pot, D P_2, to act as previously mentioned, and to suddenly cut off steam from the cylinder by closing the

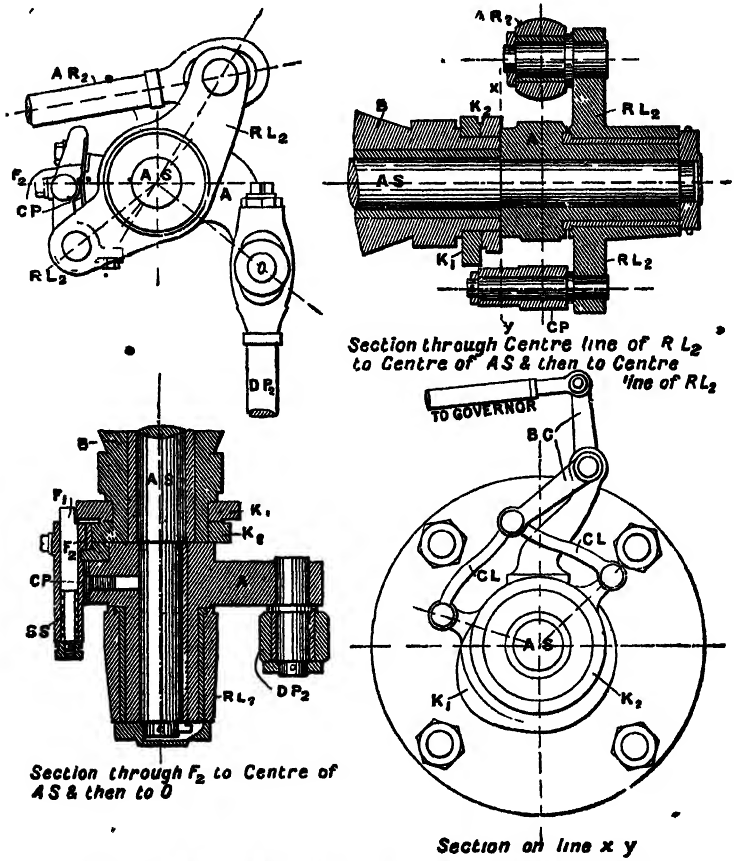


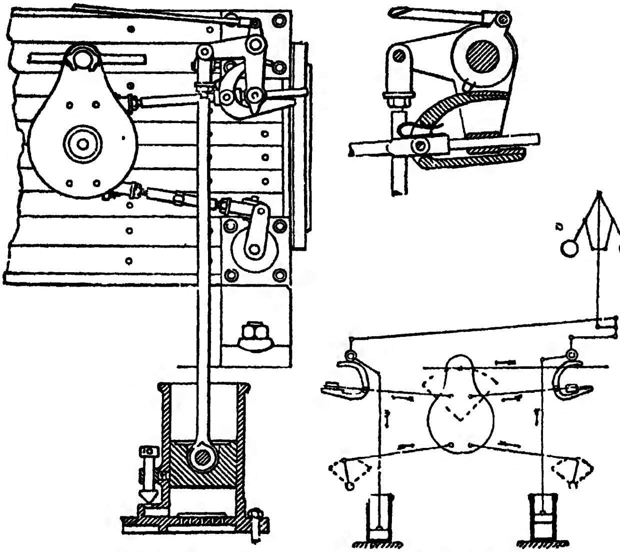
FIG. 8.—DETAILS OF FARCOT-CORLISS SINGLE ECCENTRIC TRIP VALVE GEAR.

INDEX TO PARTS.

A R_2 for Admission valve rod.	B C for Bell crank to governor.
R L_2 ,, Rocking lever.	C L ,, Cam links.
C P ,, Catch plates.	$K_{1, 2}$,, Cams.
A ,, Arm keyed as A S.	$F_{1, 2}$,, Fingers.
A S ,, Admission valve spindle.	S S ,, Spiral spring.
D P_2 ,, Dash pot.	

admission valve. During the whole time of the admission of the steam to the cylinder, the cam K_1 presses inwards the finger F_1 against the resistance of the spiral spring, S S; thus giving a longer or a shorter admission of steam until the governor acts on cam K_2, as just described.

Reynolds-Corliss Valve Gear.—Fig. 9 serves to illustrate another style of single eccentric Corliss valve gear for a horizontal steam



Dash Pot *Diagrammatic Centre Line View of Connections.*

FIG. 9.—ORDINARY SINGLE ECCENTRIC REYNOLDS-CORLISS VALVE GEAR WITH TRIP MOTION FOR A HORIZONTAL ENGINE.

engine. Index letters to the several parts of this and of the next figure have been intentionally omitted, because, after the very fully detailed descriptions which have been given, the student should exercise a little patience and perseverance in tracing out the connections and the action of this gear, without further assistance. He should sketch, letter, and describe the construction and action of this gear as an exercise in his note-book. (*See Questions at the end of this Lecture.*)

Double Eccentric Gears.*—Fig. 10 shows an outside photographic view of one form of this double eccentric gear, known

* See Appendix H for a full description of Dobson's Trip Gear re horizontal and vertical engines, and its application to Cole, Marchent & Morley's Compound Engine.

as the American Reynolds-Corliss type. On the left-hand side is seen the vertical motion rod actuated by the one eccentric and connected to one wrist-plate. From the top and the bottom corners of this wrist-plate, rods are connected to the rocking levers of the top and the bottom steam valves. In the same way the right-hand motion rod, wrist-plate, short rods, and rocking levers work the top and bottom exhaust valves. The

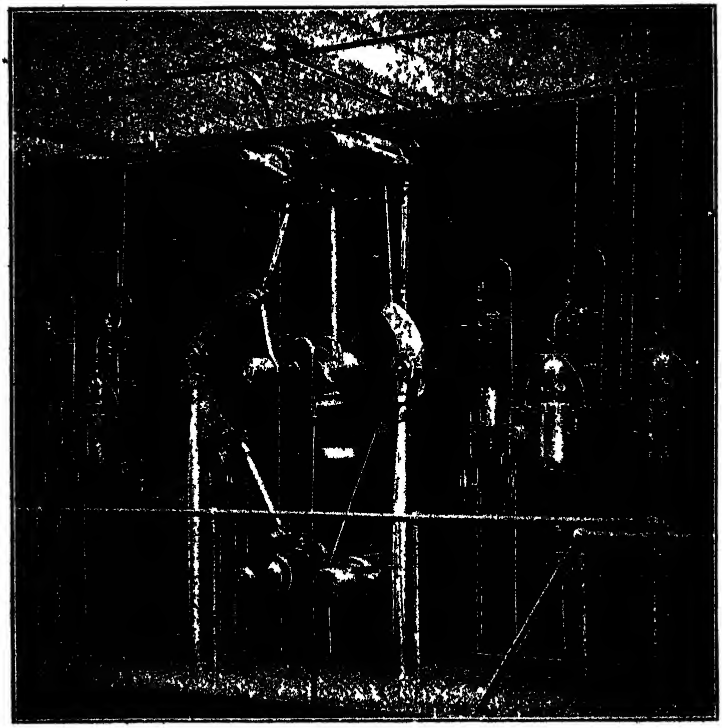


FIG. 10.—REYNOLDS-CORLISS VALVE GEAR FOR ONE CYLINDER OF THE 5,000 H.P. GLASGOW TRAMWAY ENGINES.

dash pot is situated between the lower inner sides of the wrist-plates. The complete oiling cups and pipes for the forced lubrication are also plainly seen, and this special arrangement will be again referred to in this Lecture.

The Seigrist System of Automatic Lubrication for Large Engines.—The complexity, size, and number of engines, dynamos, &c., which are now to be found in large central power stations—such as those in the Glasgow Electric Tramway Power House, at Pinkston, where over 22,000 H.P. is located in one room—have necessitated the discarding of the old system of oiling from dozens of isolated oil cups and hand-filled lubricators. Not only did the old system require a large number of men to

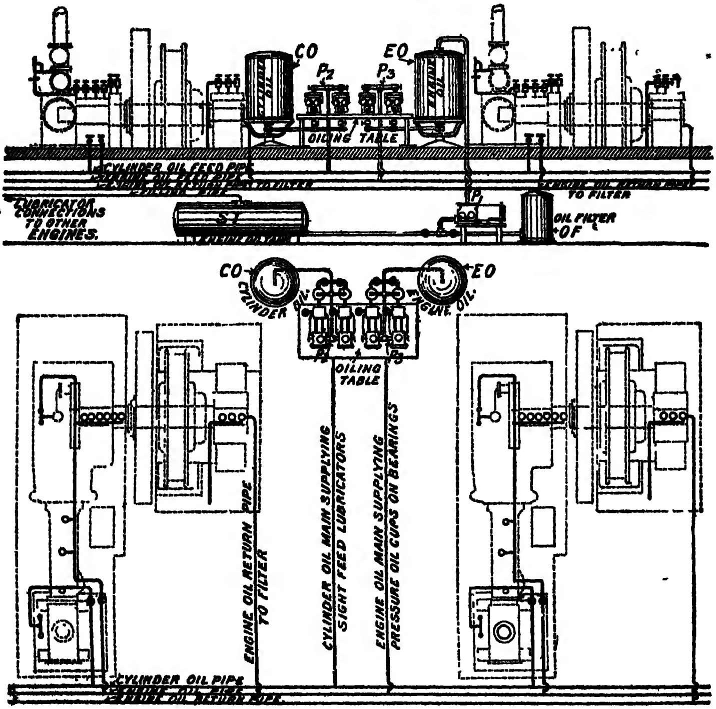


FIG. 11.—GENERAL ARRANGEMENT OF THE SEIGRIST SYSTEM OF AUTOMATIC LUBRICATION FOR LARGE ENGINES.

carry the oil from the filtering plant to the engine-room, but a considerable number had also to be constantly moving about, feeling bearings and pouring oil into the various cups, according to their respective requirements. The personal error or inattention of those so-called "travelling oil-cans," often entailed heated bearings, loss of power, and outlay for repairs, with waste of

time and money for changing over, or the laying up of one set of engines until defects were overhauled whenever the number of employees was diminished or the constant rigid system of supervision relaxed.

By the introduction of a properly installed and complete automatic system of lubrication, these shortcomings may be entirely overcome. Not only may all the valves, cylinders, slides, and bearings be thoroughly and efficiently oiled from one oiling table, but the number of attendants, their total wages bill, and the net quantity of oil used be very materially reduced.

In Pinkston Station, the oil in continuous circulation between drain pipes to filters, in filters and overhead tank was equal to 500 gallons. The oil delivered on the bearings per hour was 50 gallons per engine With the engines running twenty hours per day and seven weekly working days, the oil circulated to and from each engine was about 7,000 gallons per week; or, with a weekly make up of 40 gallons of oil, we see that only $\frac{4}{7}$ of a gallon was lost per 100 gallons used by the machinery. No returns or pipes were used from the valve gear, pump-room plant, or any auxiliary plant. The total savings in cylinder engine oil and waste amounted to over £40 per week when compared with what it was before the new automatic system of lubrication was adopted.

Manipulation of the Oil.—The oil is brought alongside the power-house in a railway truck oil tank or in an oil cart. It is then passed through the supply or filling pipe into the storage tank, S T. This tank is in connection with the suction inlet of a duplex direct-acting steam pump, P_1, by which the oil is forced through the piping to the engine oil reservoir, E O. This steam pump, P_1, is also connected to the oil filter, O F. Cocks are provided on the two sets of pipes, so that, by closing one or other of these valves, the pump can take oil either from storage tank, S T, or filter, O F. The oil from the reservoir, E O, is led into another set of pumps, P_3, arranged upon a metal table at a convenient level above the engine-room floor. This pump, P_3, discharges the oil into a system of piping carried throughout the engine-room and communicating with each moving part requiring lubrication. The waste oil from each part is caught and led into a central position, from which it is carried by piping to the oil filter, O F. After the impurities are removed by the filter, the oil is again pumped into the tank, E O, to be used as before for the bearings.

The special cylinder oil from the reservoir, C O, is led by piping to the steam pump, P_2. The oil is forced by this pump along pipes to the engine cylinders.

Each pump is provided with pressure gauges, spring-loaded valves, and an automatic governor, by means of which the speed of the pump is controlled, according to the quantity of oil required by the engines. Special fittings are bolted to the cylinders to permit of the most minute adjustment in the oil supply.

Cylinder Lubricator.—Two views are given, showing the construction of the cylinder lubricator of the Seigrist system. The space under the diaphragm, D, is in communication with the cylinder of the engine, while the space above the diaphragm

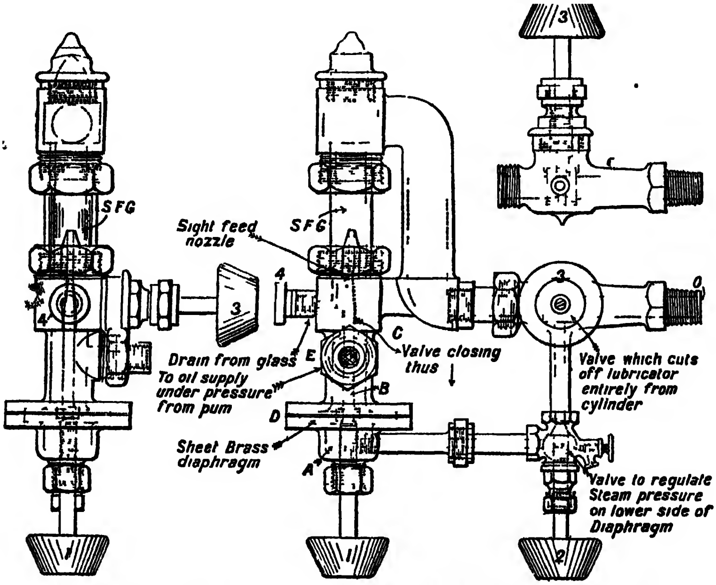


FIG. 12.—SIGHT FEED LUBRICATOR FOR THE SEIGRIST SYSTEM OF AUTOMATIC LUBRICATION.

is in communication with the oil supply under pressure from the automatic pumps, P_2 (*see General Arrangement*). By raising or lowering the adjusting handle 1 or 2, the amount of oil passing the valve can be adjusted to the requirements of the engine cylinder.* This oil flows upwards through the water in the sight-feed glass tube, S F G, in the same manner as for ordinary sight-feed lubricators. The oil, after passing through the outlet O, is caught by the flow of steam and carried into the cylinder. It

will be seen from these two views, that if the engine should be stopped, that the pressure of the oil supply from the pumps, P_2, will force the valve down on its seat, because the steam pressure on the underside of the diaphragm, D, is removed, thus preventing any waste of oil while the engine is standing.

Compound Engine with Automatic Lubrication.—A good example of the lubrication of several journals and slide blocks from one common source of supply under pressure, is furnished by Belliss and Morcom's compound engines for the direct driving of dynamos. It will be seen from the Folding Plate, that not only the main crank-shaft bearings, but also the crank-pins, slide-blocks, the upper ends of the connecting-rods, the piston-valve eccentric and its rods, are all supplied with oil from a small pump worked by the same eccentric which moves the piston valve. The oil is thereby forced through *each* bearing under a pressure of 10 lbs. per square inch, and is again and again sent on its soothing mission for months at a time, without change or great loss in quantity. A heavy lubricating oil is used, and it always returns to the small pump through a filter which removes any grit that it may have picked up from the bearings. This is a very different state of matters from the old "travelling oil-can" system, when the quantity of oil applied and the times of application were as erratic as the judgment of the attendant.

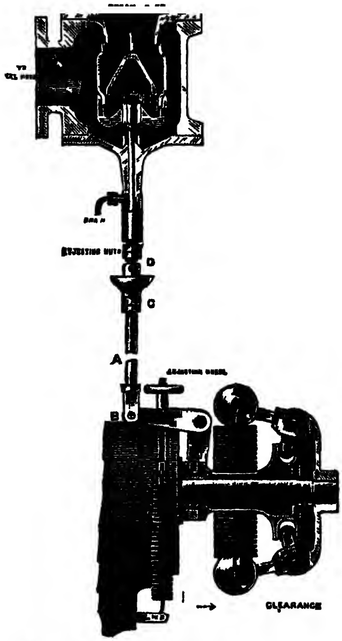


FIG. 13.—GOVERNOR FOR BELLISS MORCOM ENGINES.

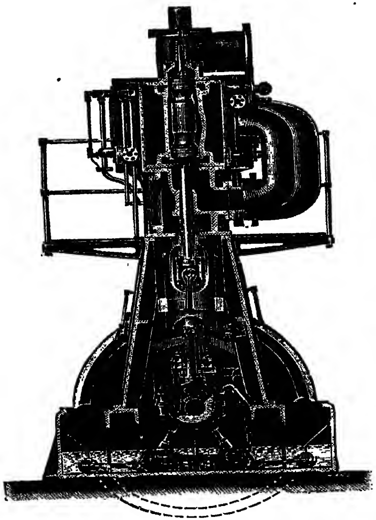
A special feature of this engine is that both cylinders are supplied with steam by one slide valve, worked by one eccentric and one valve rod. The cranks are set opposite to each other, and the steam is admitted simultaneously to the top of one cylinder and the

bottom of the other. By this arrangement, the reciprocating parts are to a great extent balanced, the stresses on the bearings are much reduced, and a high speed of revolution is possible without setting up undue vibration.

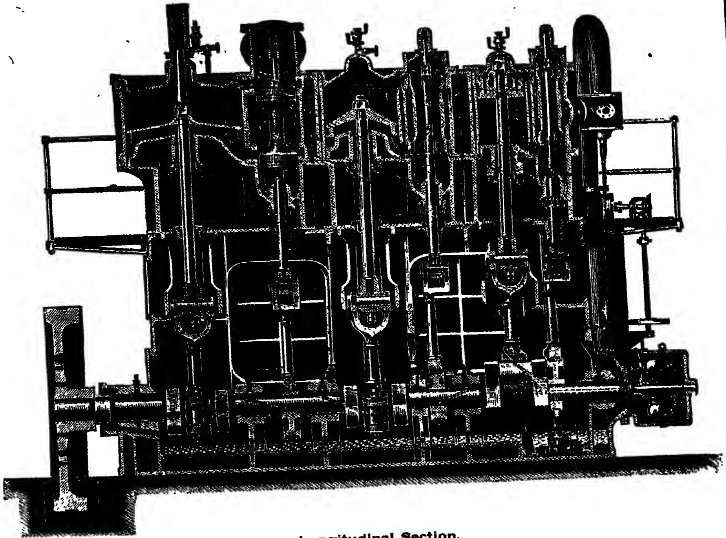
The engine is fitted with a centrifugal governor (Fig. 13), carried on the crank shaft, and connected to an equilibrium throttle valve. The governor gear is arranged so that the speed of the engine is capable of being altered by the adjusting wheel through a wide range of variation whilst running. The centrifugal force of the two governor balls is chiefly resisted by the springs connecting them, but is partly opposed by the adjusting spring. In the event of any one of these springs breaking the balls instantly fly outwards, thus closing the throttle valve and stopping the engine. A variation of speed not exceeding 3 per cent. between full and no load can be guaranteed with these engines, and consequently they are found suitable for the direct driving of dynamos supplying current to an electric light or power installation, as shown by the Folding Plate.

Triple-Expansion Engine with Automatic Lubrication.*—The accompanying Folding Plate serves to illustrate one of the best examples of vertical, inverted cylinder double-acting quick-revolution engines. These engines run at very high speeds without any fear of excessive wear and knocking of the connecting-rod brasses. The difficulty usually experienced when running double-acting engines at over 300 revolutions per minute arises from the necessity of such close adjustment of the brasses to avoid audible knock and shock. Thus, a small increase in temperature of the crank-pin may cause sufficient expansion to make it overtake the small clearance which is generally allowed for the bearing when cold. Consequently, this close adjustment renders the pin and its bearing liable to get hot and seize. The success of the Belliss & Morcom engines is largely due to supplying the lubricant under pressure to the several moving parts. The pressure necessary for this purpose is not nearly equal to the maximum pressure on the bearing due to the weight of the engine, but only sufficient to force the oil into the bearing during a return stroke. The time taken by the piston in its upward stroke is too short to allow the oil to be squeezed from between the rubbing surfaces before

* Students are referred to the following papers should they desire any further information upon this subject:—*Proceedings of the Inst. C.E.*, vol. cxxxvi., "High-speed Engines," by John Handsley Dales, A.M., Inst. C.E.; and vol. cxlv., "Delannay-Belleville's High-speed Engine," by M. Aliamet.



Vertical Cross Section.



Longitudinal Section.

Self-Lubricating Triple-Expansion Engine

BY BELLISS & MORCOM LTD., ENGINEERS, BIRMINGHAM.

For Description and Tests, see Professor Jamieson's Text Book on "Steam and Steam Engines.

the pressure is again reversed and a fresh supply of oil given to the surfaces, as just explained in the case of the compound engine by the same makers.

Results with Superheated Steam.—The following set of results, plotted in Fig. 14, were obtained from a 300 B.H.P. triple-expansion condensing engine using different degrees of superheat up to 307° F., or a total temperature of 677° F. The results show that the percentage gain or saving in pounds of steam per I.H.P.-hour agree very closely with those quoted in Lecture XVI. with reference to the Willans engine. These results are quoted in the following table, as well as others from engines of different powers by the same makers, working at different loads and speeds The student should plot out these

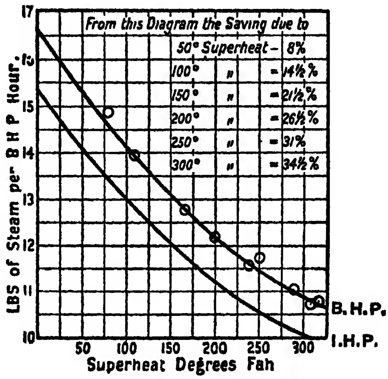


FIG. 14.—RESULTS OBTAINED WITH A 300 B H P. BELLISS & MORCOM'S TRIPLE-EXPANSION ENGINE, USING SUPERHEATED STEAM OF 160 LBS. PRESSURE, AND A VACUUM OF 26·75 INCHES AT 475 REVOLUTIONS PER MINUTE.

results to scale, and thus present their several values in graphic form.

Although the remarkable economy shown by the results plotted in Fig. 14, of requiring only 10 lbs. of steam per I.H.P.-hour, were obtained from these quick revolution engines, yet, the author feels bound to state, that great care should be observed by those who meditate using such highly superheated steam of 600° F. or more.

Some Results obtained with Belliss & Morcom's Triple-Expansion Self-Lubricating Engines, using Superheated Steam.

Data.	D.E. 475 R.P.M. 25/1/04.	Dynamo Engine at 340 R.P.M. 9/2/03.				D.E. 200 R.P.M. 9/8/03.	Dynamo Engine at 380 R.P.M. 24/10/02.			Dynamo Engine at 360 R.P.M. 2/12/02.			
Load,	Full	Full	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	Full	Full	Full	Full	Full	Full	Full	Full
Steam pressure in range,	160	165	163	164	155	183	170	170	170	190	190	190	190
Temperature in Fah. degrees,	677°	572°	565°	553°	472°	445°	485°	511°	559°	407°	432°	472°	492°
Superheat in Fah. degrees,	307°	200°	193°	180°	104°	64·4°	110°	136°	184°	24°	49°	89°	109°
Vacuum at engine,	26·75″	25·4″	25·9″	25·8″	26″	25·6″	26″	26″	26″	25″	25″	25″	25″
B.H.P.,	300	522	394·5	262·5	130	...	...	...	...	...	...	...	...
Consumption per B.P.H. hour,	10·75	11·15	11·23	11·6	14·35	13·3	13·3	13	11·9	15·4	14·6	13·6	13·2
Kilowatts,	...	...	...	...	...	1,436	216	220·4	220	305	308	306	308
Consumption per K.W.T. hour,	15·45	16·1	16·5	17·7	22·7	19·15	19·23	18·78	17·21	22·2	21·1	19·6	19
Dynamo efficiency (assumed),	93 %	93 %	91 %	88 %	85 %	93 %	93 %	93 %	93 %	93 %	93 %	93 %	93 %

Necessary Precautions to be observed with Superheaters and with Highly Superheated Steam.—1. Superheater tubes are liable to get warped, burned, or chemically acted upon unless properly designed, made, erected, and worked.

2. Highly superheated steam erodes or cuts into brass and gun metal. Nothing less than nickel steel would permanently stand its effects upon valves and valve seats.

3. Highly superheated steam spoils the working surface of the softer kinds of cast-iron cylinders. Great care should be taken in applying superheated steam to cylinders which are not made of the very best, hard grey, close-grained cast iron.

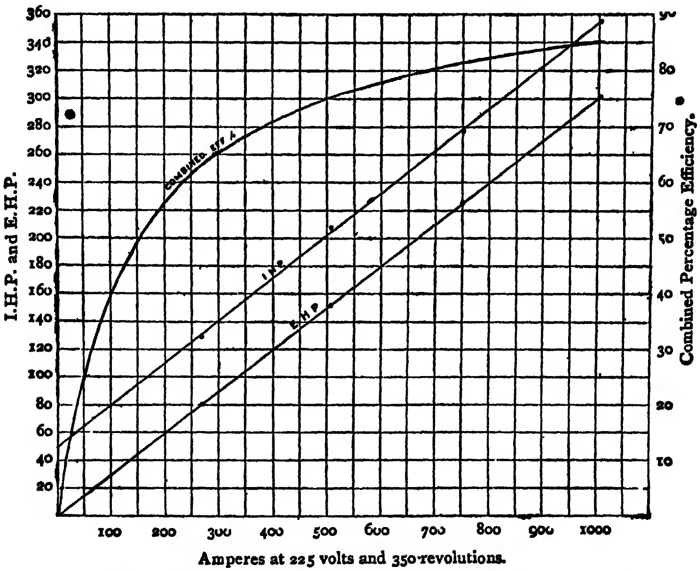


FIG. 15.—EFFICIENCY CURVE FROM A FULL LOAD TEST MADE ON A 360-H.P. NON-CONDENSING WILLANS ENGINE AND A SIEMENS DYNAMO.

DATA.—I.H.P. = 355·7; E.H.P. = 302·2; mean steam pressure = 51·17; revolutions per minute = 350; cut-off = ·5; steam pressure = 136 lbs. per square inch.

4. When plumbago or graphite is used as a lubricant for cylinders, it is apt to clog and jam the piston rings, &c.

5. It has been found that engines in a first-rate condition

may be run with very little lubrication. When lubrication is necessary with superheated steam, then only the best kind of high flash point lubricant should be used, such as "valvoline."

6. Steam pipe and cylinder laggings, as well as everything which come into contact with steam pipes containing very highly superheated steam, should be fire-proof, since they may be subjected to temperatures approaching 700° F.

7. The stresses arising from highly-superheated steam were very great, and due allowance must, therefore, be made in the design of an engine to permit of free expansion without twisting, warping, or overstraining the parts thus affected by the extra heat.

Tests of Willans' Engine.—The foregoing curves (Fig. 15), together with the attached data, give a clear idea of the combined efficiency of the indicated horse-powers, I.H.P., the electrical horse-powers, E.H.P., and the combined efficiency of a 360 H.P. non-condensing compound Willans engine when coupled to a Siemens dynamo.

The following data gives the mean results of four sets of independent tests of a 400-H.P. Willans triple-expansion condensing engine when supplied with ordinary dry saturated steam:—

Test of Willans' Engine.

Effective area of cylinders—three of each: high pressure, 90·836 *square inches; intermediate,* 345·44 *square inches; low pressure,* 587·175 *square inches. Stroke,* 10·24 *inches.*

Mean boiler pressure above the atmosphere, .	190 lbs. per sq. in.
Mean admission high-pressure cylinder, . .	169·5 lbs. per sq. in
Mean effective pressure on low-pressure cylinder,	28·918 lbs. per sq. in.
Mean vacuum, ...	25·925 inches.
Mean revolutions per minute, ...	299·8.
Mean I.H.P., ...	394·775.
Mean total feed-water per hour, ...	5,050 lbs.
Mean deductions for separator and other drains per hour, ...	117·1 lbs.
Mean total steam to engine per hour, ...	4932·9 lbs.
Mean steam used per I.H.P. per hour, . . .	12·49 lbs.

Percentage Gain in Steam and in B.T.U. with Willans' Engine when supplied with Superheated Steam.—The curves and table of data given in Lecture XVI. regarding these gains show that with a triple-expansion engine giving 315 I.H.P., and using steam of 162 lbs. pressure by gauge with 180° F. of superheat, that the gain in steam used is fully 24 per cent. Now, applying

this to the previous case, where the steam used per I.H.P.-hour was 12·49 lbs. with saturated steam, we get—

$$\therefore \quad 100 \text{ per cent.} : 76 \text{ per cent} :: 12{\cdot}49 \text{ lbs.} : x \text{ lbs.}$$

$$x = 9{\cdot}5 \text{ lbs.}$$

with the same degree of superheat and assuming the same efficiency.

The Willans Central-Valve Engine.*—As will be seen from the accompanying figure the engine is single-acting, having all its brasses and moving parts constantly in compression, to enable it to run at a high speed without knocking. The slide valves are of the piston type, and work inside the piston-rods. This method affords a very direct distribution of the live steam and a free drainage for the condensed steam. The high piston speed employed in this engine is in itself conducive to economy, and the Willans engine (as proved by many tests of undoubted authority) is one of the most economical steam motors. With a small condensing engine indicating only 20 horse-power and running at 400 revolutions per minute, a consumption of 13 lbs. of steam per I.H.P. hour has been recorded, and a little over 18 lbs. when worked as a non-condensing engine.

Cranks, Connecting-rods, and Eccentrics.—Each line of pistons is connected to its corresponding crank by two exactly similar connecting-rods, with a space between, in which works an eccentric, *forged solid upon the crank-pin*. The connecting-rods at their top end engage two hardened steel pins, so supported that the pressure of the rods exerts no twisting stress upon them, and the eccentric-rod plays up and down freely in the space between them. The piston slide valves move *inside the hollow piston-rod*, R, which passes completely through the line of pistons, and through the ends of the cylinders. The reason for placing the eccentric on the crank-pin, and not on the crank-shaft as usual is, that the valve face (*i.e.*, the inside surface of the hollow piston-rod) *moves with the pistons*. Consequently, the valve-motion required is a motion *relative to the pistons*, and this is obtained by mounting the eccentric on the crank-pin, which, like the piston-rod, moves up and down with the pistons. Though its lead is set out differently from that of an ordinary eccentric, its effect upon the movement of the valves is exactly the same.

Cylinders.—The annexed sectional view shows a standard pattern engine of G.G. size, which has low-pressure cylinders of 14″ diameter and 6″ stroke.

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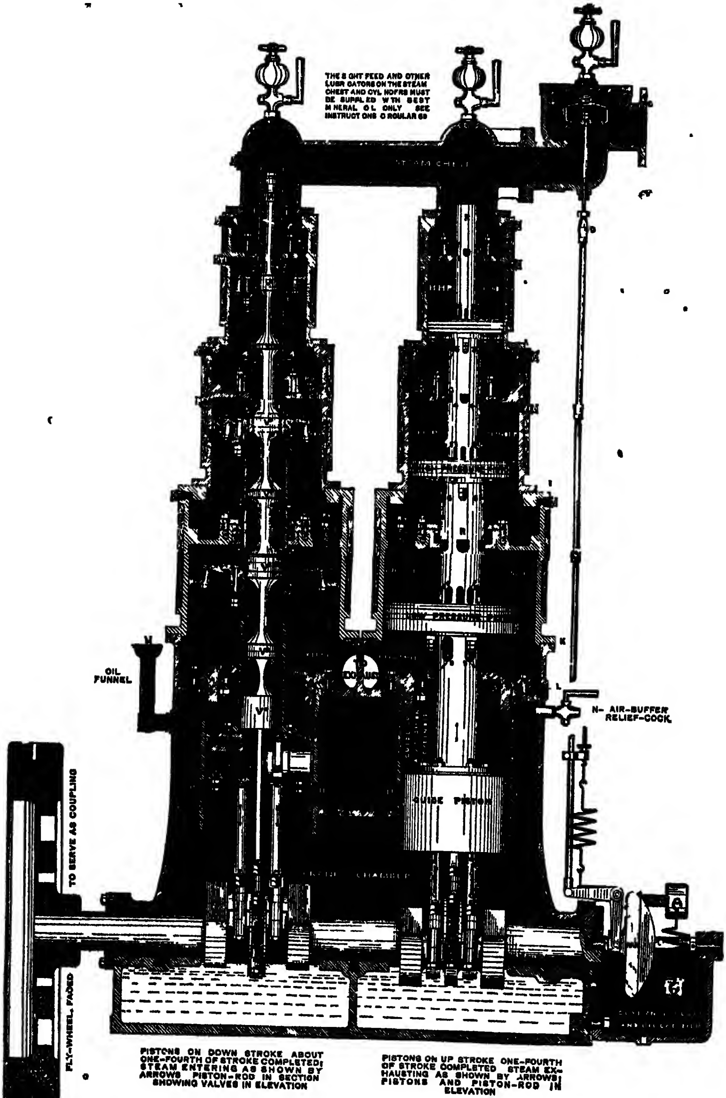
* Students who desire to thoroughly study the excellent work done by Mr. Willans in the evolution of quick-revolution engines, should refer to the following papers with the discussions upon them. They should also refer back to Lecture XVI. for the saving in feed-water, steam, and B.T.U. See *Proc. of Inst. C.E.*, vol. lxxxiii., p. 106, for "High-Speed Motors," by John Imray, M.A., M.Inst.C.E.; vol. xciii., p. 128, for "Economy Trials of a Non-condensing Steam Engine: Simple, Compound, and Triple;" vol. cxiv., p. 2, for "Steam Engine Trials;" and vol. xcvi., p 230, for "Economy Trials of a Non-condensing Steam Engine: Simple, Compound, and Triple"—all by P. W. Willans, M.Inst.C.E.



WILLANS' CENTRAL-VALVE TRIPLE EXPANSION ENGINE.

The right-hand line of pistons, with the hollow piston-rod, is shown in elevation, while the left-hand line is in section, with the piston-valves in elevation. In the right-hand line of pistons (which is upon the up-stroke) the course of the exhaust steam is indicated by arrows, but the piston-valves are necessarily invisible.

It will be noticed that a compound, or even a triple-expansion, Willans engine, may be run as a simple engine without removing the upper cylinders, by merely removing the upper pistons, and the piston-valves corresponding with them. In fact, either the steam-pistons or the valves might be left, were it not for the useless friction of the rings. The upper cylinders, in such a case, become a mere extension of the steam chest. It is sometimes a practical convenience to be able thus easily to alter a compound to a simple engine, or a triple-expansion to a compound.

Steam Distribution.—Referring to the left-hand line of pistons (which has completed $\frac{1}{4}$ of the down-stroke), it will be seen that the steam from the boiler (after it passes the throttle-valve and the steam-chest) enters the hollow piston-rod by the uppermost openings, 1, 1. From thence it goes into the H.H.P. cylinder by the second set of piston-rod openings, 2, 2. Cut-off takes place at about ·6 of the piston's stroke by the passage of the ports, 1, 1, into the first gland, G. Exhaust takes place by the second piston-valve, V^2, rising above the ports, 2, 2, and thus permitting the steam to pass out of the uppermost cylinder through these ports, 2, 2, into the hollow piston-rod and from there through the openings, 3, 3, into the receiver for the next cylinder. From this receiver, the steam again enters the hollow piston-rod by the ports, 4, 4, and out to and above the H.P. or second piston by the openings, 5, 5. Cut-off and exhaust take place for this in the same way as for the first cylinder—viz., cut-off by the passage of ports, 4, 4, into the second gland, G, and exhaust by the piston-valve, V^6, rising above the ports, 5, 5, thus permitting the outgoing steam to pass through them and then through the openings, 6, 6, into the receiver for the third or low-pressure cylinder. Here again admission, cut-off, and exhaust take place, as in the case of the previous two cylinders, viz., admission from the second receiver by ports, 7, 7, and 8, 8; cut-off by ports, 7, 7, becoming covered in their downward passage by the third gland, G, and exhaust by piston-valve, V^8, rising above ports, 8, 8, and letting the steam through them into the hollow piston-rod and out through holes, 9, 9, into the exhaust chamber during the whole of the up-stroke. The exhaust pipe from this chamber may either communicate directly with the atmosphere or with a condenser. Piston-valves, V^9 and V^{10}, constitute a guide for the bottom of the valve-rod, V^{10}, has no packing, and there are holes in it in order to afford a free passage of water or oil through the same.

It will be noticed that in the simple non-condensing engine, the steam remains in the engine, from the commencement of admission to the end of exhaust, for one revolution, as in ordinary engines. But in the compound non-condensing engine, the steam remains for two, and in the triple expansion engine for three, whole revolutions. In other words, the steam is practically quiescent in a receiver of some kind for half a revolution between each two stages of expansion, and this (which is only possible in a single-acting engine) enables the range of temperature in the several stages to be divided advantageously.

Drainage.—The water above each piston is swept downwards by the exhausting steam into the space below *during the whole of the exhaust stroke;* it has not to be carried by the piston to the top of the cylinder, and then driven out suddenly through the port in a more or less upward

direction, as in the case in other forms of vertical engine. The Willans engine has, therefore, unique advantages in getting rid of water from the cylinders, apart from the action of the relief-valves.

Air Cushioning in Guide Cylinders.—Reference has been made to the fact that all the moving parts are constantly in compression—a condition rendered possible only by the fact that the pistons are single-acting, giving no pull to the crank upon the up-stroke, but only a push upon the down-stroke. In any engine running at high speed the moving parts can only be kept in compression upon the up stroke by very great cushioning. This is rarely obtained in other high-speed engines without excessive compression in the cylinders, which naturally involves a certain waste of steam. Sometimes, when a high-speed engine exhausts into a vacuum, sufficient cushion cannot be obtained *at all* by the usual means. In the Willans engine very little compression is given in the steam cylinders, for little or none is required. The requisite cushioning is obtained independently by the guide pistons. These pistons, on the up-stroke, compress the the *air* contained in the guide cylinders, and thus any desired amount of cushion can be obtained, according to the clearance allowed.* The work expended in compressing the air *is given out again by its expansion on the succeeding down stroke*, and the loss, when the engine is running at a good speed, is proved by indicator diagrams to be too minute to be worth consideration. There are holes, 11, 11, in the guide cylinders, which are uncovered by the guides at the bottom of the stroke. As the casing or chamber which surrounds the guide cylinders (and which forms part of the framing of the engine) is open to the atmosphere, it is evident that the air compression always commences at atmospheric pressure, and is constant and invariable in its results, whatever alteration may be made in the pressure of the exhaust steam.

The Brasses and all parts in Compression.—The upper crank-pin brasses of the connecting-rods are wider than the lower ones. This is because the upper brasses alone are intended to be in actual contact with the crank-pins; the lower ones are only a stand-by in case of accident. All the moving parts of the engine are designed to be strictly in "constant thrust;' the connecting-rods are *always in compression, never in tension.* A small hole is drilled in each guide piston, $\frac{1}{8}$ inch in diameter, so as to be just visible below the bottom edge of the guide cylinder when the crank-chamber door is removed, and when the piston is at the bottom of its stroke. When the entire diameter of this small hole is in view below the guide cylinder, it is time both to set up the brasses (so as to reduce the play) and to pack up the connecting-rods by inserting packing pieces between the big ends of the connecting-rods and the brasses. The connecting-rods, however, must not be packed up sufficiently to take the hole quite out of sight, for its lower side must still be in sight (at bottom stroke) under the edge of the guide cylinder. If the hole goes out of sight entirely, there will not be enough clearance for safety between the low-pressure piston and the top of its cylinder.

The eccentric-rod is also intended to work in compression, in the same way as the connecting-rods. The holding-down pressure is furnished by the steam in the steam chest, acting constantly upon the uppermost piston-valve, V^1. It may sometimes happen, if the engine is run with a very light

* The amount of cushion is fixed in each case to suit the intended speed, and may be insufficient to prevent knocking if that speed is largely exceeded. If for any reason it is desired to run an engine materially faster than was originally intended, and the engine is found to knock at the increased speed, the speed must be reduced until the knocking disappears.

load, but at a high speed, that the pressure in the steam chest (being much throttled down by the governor) is insufficient to keep the eccentric-rod in contact with the eccentric upon the up-stroke. If this be the case then a slight knocking may be heard, as the lower eccentric-strap is purposely left an easy fit upon the eccentric. Such knocking is unimportant, if not allowed to continue too long, and it will cease as soon as the engine is given work to do.*

A further reason for the moderate wear of the brasses (and eccentric straps) is that they dip bodily into the lubricant in the crank chamber at every revolution. In doing so they splash it over the main bearings, and to the upper ends of the connecting-rods and eccentric-rods, and into the guide cylinders, as well as into that part of the hollow piston-rod where the guide, V^{10}, works. The lubrication of the working parts (other than steam pistons and valves) is thus completely automatic. It is sufficient to mention here that (according to the usual method of working) the crank chamber contains not oil only, but oil and water mixed. As the temperature of the mixture cannot possibly rise above that of boiling water, there is a practical guarantee against hot bearings, so long as the supply of water is maintained and suitable oil is used.

Internal Relief-Valves.—In the low-pressure cylinders of all engines, and in the high-pressure cylinders, if large enough to be so treated, internal relief-valves are fitted, consisting of a gun-metal plug screwed into the top of the low-pressure cylinder. The plug is pierced by holes, covered by a single thin gun-metal disc. When the disc is raised, there is free communication between the cylinder and the receiver (or steam chest) above it. It is kept down under ordinary circumstances by the excess of the receiver-pressure over that in the cylinder; therefore no spring is required, and there is no part liable to get out of order. If from water in the cylinder, or any other cause, the pressure rises above that in the receiver, the valve lifts, and though the water is only passed back into the receiver, the relief is found to be sufficient, and, in fact, far more effective than that given by ordinary external relief-valves. Engines so fitted have been tested by discharging a cubic foot of water suddenly into the steam-pipe; also by connecting the steam-pipe with the water-space of the boiler (by a ½-inch pipe, with a difference of 80 lbs. between the pressure in the boiler and that in the steam-pipe) without auy injury to the engine in either case. In cases where internal relief-valves cannot be used, ordinary external valves are fitted. When an engine is run without load the compression in the low-pressure cylinder may rise beyond the pressure in the receiver; the disc of the valve may then be heard to lift at each revolution, but the noise will go off as soon as the receiver-pressure is increased by giving the engine work to do.

Air-Cocks.—Air buffer relief-cocks, N, are fitted upon the guide cylinders, in order to avoid compressing the air in them when the engine is being turned by hand, and to facilitate starting. If the cocks are opened at starting, they must be closed as soon as the engine gets fairly under-way. They must never be open when the engine is running at full speed, or the necessary cushion will be wanting.

Drain-Cocks.—The drain-cocks on the receivers should be fully opened before starting, and should be kept open for a short time after starting They must, however, be closed and be kept closed while running, except occasionally to draw off any water which may have collected—when they

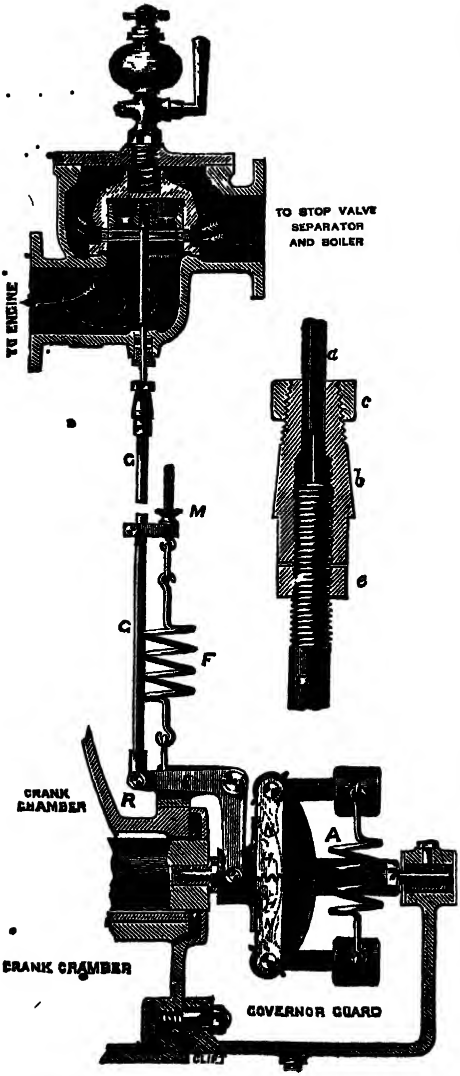
* The principle of working with all brasses "in constant thrust" is of the utmost importance and value, and is the primary cause, not only of the silent running of the Willans engine, but of the almost complete absence of wear in the brasses.

should be opened only to an extent just sufficient to allow the water to escape. The receiver-drains are connected, by copper pipes, with the exhaust chamber.

Lubrication.—Usually only one lubricator is required, of the "sight-feed" pattern. After the engines have worked for some time, a very small supply of *good* mineral oil will suffice for the cylinders. At M, on the sectional view, is shown a funnel for introducing oil into the crank chamber —preferably the best castor oil, but *not* mineral oil. The funnel (the usual place for which is on the front of the engine, and not where shown) also establishes communication between the air-cushion cylinders and the atmosphere, when the guide pistons are at the lower end of their stroke. When water is required to be added, it should be poured in through the open top of the lubricant gauge, and *not* through this oil-funnel. A gauge enables the quantity of lubricant in the crank chamber to be easily ascertained at any time by the attendant. The normal height at which the lubricant should be maintained is from three-quarters of an inch to an inch below the underside of the crank shaft.

Separator.—Every engine is now fitted with a steam dryer, or separator, mounted on one corner of the bed-plate The steam enters at the top and descends through a hanging pipe into the body of the separator. After leaving the pipe it turns upwards to the exit, which is near the top, while the particles of water, which are heavier, are shot downwards by the velocity with which they leave the hanging pipe A gauge-glass is fitted, and a drain, the cock on which should be so adjusted as to keep a little water in the glass, just in sight.

Governor.—1. In the accompanying figures the governor balls are shown in the position they assume when controlling the engine. The throttle-valve is of the piston type without rings, and it works up and down in a bush with a closed top. The bush is held down by a coiled spring above, and by the steam pressure, and its lower end, which is faced, makes a steam-tight joint against a face on the casing, as shown. The boiler steam is admitted from the outside of the bush, in which there are two rings of ports, the amount of opening of the lower ring being regulated by the position of the lower edge of the throttle-valve. Corresponding with the upper ring of ports is an annular port in the throttle-valve; the distance of this from the lower edge of the throttle-valve being such that the upper ports commence to open slightly earlier than the lower ones. In cases where, owing to low-pressure, or other causes, only a very small drop in pressure can be allowed between the steam-pipe and the steam-chest, a third ring of ports is sometimes added, with a second annular port in the throttle-valve. The lubrication of the valve is effected by passages in the body of the bush, supplied from a grease cup on the cover. The sight-feed lubricator is attached to the boss dotted on the engraving; it delivers oil on the engine side of the throttle-valve. The spring, F (the lower part of which is hooked to a fixed point on the bracket which supports the bell-crank, E), maintains a constant down pull on the rod, G, and so tends to close the valve. It also tends to force the balls further apart, by depressing the end, K, of the bell-crank, E, and so pushing outwards the loose collar, C (shown partially dotted), and the short ends, N, N, of the arms which carry the balls. But the pre-arranged relation between the centrifugal force of the balls at different speeds, and the pull of the springs, A, at different lengths is such, that so long as the engine is running even slightly below its speed, the pull of the springs, A, overcomes both the centrifugal force of the balls and the pull of the spring, F, and the balls remain near together,



Centrifugal Governor for Willans' Engine.

and the valve in its widest open position. At the intended speed (or slightly below it) the centrifugal force of the balls, helped by the spring, F, causes them to overpower the springs, A; and as the arms, N, N, move outwards, and permit the collar, C, and the bell-crank, E, to follow them. The spring, F, is, therefore, able to draw the rod, G, downwards, and to close the valve more or less completely, until the engine runs at its normal speed. If the speed, for any reason, such as reduction of load or increase of boiler pressure, begins to exceed that intended, the balls fly open and the throttle-valve closes until the speed falls again. If, on the other hand, the speed diminishes, the balls are drawn together by the springs, A, and approach one another, the spring, F, is over-powered; and the valve opens. In order to permit a certain amount of end play in the crank-shaft, without causing movement in the throttle-valve, the governor spindle fits loosely in a corresponding hole in the shaft, and is free to move a short distance endways. The outer end of the spindle is supported in a bearing formed in the governor guard, and carries a phosphor bronze ring, which works against the face of the bearing. The spindle is kept up

against this face by the tension of the spring, F, acting through the bell-cranks, E, unaffected by end-play in the crank-shaft which drives it. The levers, H, H, are rigidly attached to the arms which carry the balls, and their free extremities are geared together, as shown. By this means the governor is balanced against gravity in all positions.

It will be noticed that there are four elements which determine the action of the governor in controlling the engine, viz.:—

(1) The weight of the balls which measures their tendency to fly apart at any given speed. If the balls are made heavier, they will overpower the springs, A, at a lower speed; if lighter, the engine must run faster before they will come into action.

(2) The pull of the spring, A, against the balls, and the ratio in which it varies as the distance between them alters.

(3) The pull of the spring, F, assisting the balls.

(4) The position of the valve relatively to the balls, as determined by the adjustment of the length of the spindle, G.

2. The spring, F, as has been explained, assists the balls to open, though its action is small in comparison with the centrifugal effect of the balls. If more tension is put upon it, by means of the thumb-nut, M, the balls will open at a lower speed; if the tension is reduced, they will not open until a higher speed is reached. A moderate adjustment, therefore, can be given by the nut, M. In cases where a considerable range of speed is required a different method is adopted.

Criticism of the Farcot-Corliss Cylinder and Position of the Valves, as shown by Figs. 1 and 2 in this Lecture.—*Best Up-to-date Corliss Cylinders and how they are Arranged.*—Figs. 1 and 2 serve the purpose of enabling the student to obtain a good idea of a Corliss cylinder with the separate positions of its four separate valves. But, first-class makers of Corliss engines aim at designing their cylinders, so that the interior of the cylinder, as well as the piston and piston-rod, may be inspected and the last two withdrawn from the cylinder as easily as possible—*i.e.*, by simply removing the cylinder covers.

Separate Parts constituting the Cylinder:—

(1) The cylinder barrel with its jacket and liner.

(2) The separate steam- and exhaust-valve chambers with ports for the front end.

(3) The separate steam- and exhaust-valve chambers with ports for the back end.

(4) The front and back end covers.

The first three of these sets of parts are so designed, machined, jointed, and bolted together, that they constitute the whole of the cylinder proper. Each part is separately machined, and may be separately repaired and adjusted.

Easy Examination of the Inside of Cylinder and the Piston.—If it be desired to examine the piston or the interior of the cylinder from either or from both ends, it is only necessary to take off and withdraw the front cover as far as the recess in the front framing will permit, and to remove the back-end cylinder cover, without touching any of the valve-chambers, steam or exhaust pipe-joints. By driving out the cotter connecting the crosshead to the front end of piston-rod, the piston with its rod may be then withdrawn from the open back end of the cylinder.

Best Position for Steam and Exhaust Valves.—The front and the back steam-valve chambers should be placed fair above their respective ends, and their short-port openings lead fair down to the very ends of the cylinder clearance spaces. The front and the back exhaust-valve chambers should be placed fair beneath their respective ends, and their short-port openings lead fair down from the very ends of the cylinder clearance spaces. The cylindrical surfaces of each of these rocking valves should work quite clear of the bore of the cylinder.

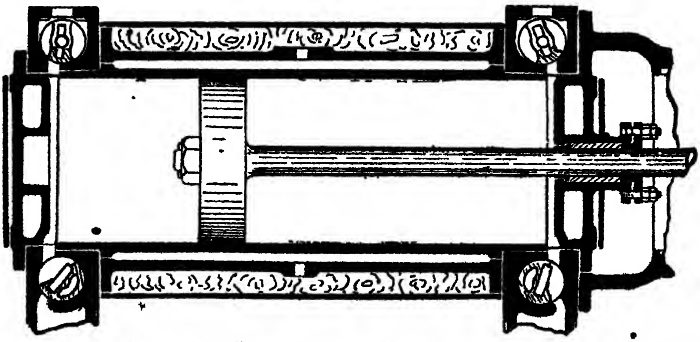
Steel Springs for Aiding Corliss Valves to be kept Tight.—Some makers of Corliss engines assist the steam pressure by aid of springs attached to the valve spindles with the object of still further ensuring the prevention of the leakage of steam past the valve faces.

Steam Jacket.—Provision should be made for keeping the jacket solely filled with dry hot steam when the engine is at work. It should therefore be thoroughly drained of any condensed steam by being connected at its lowest point with an efficient steam trap. This trap should empty into a pipe leading to the hot-well or boiler feed-pump suction chamber, as explained in connection with the illustrations and description of the S.S. "Inchdune's" engines (see Index).

Defects of the Farcot-Corliss Cylinder and Position of its Valves.—To inspect the interior of this cylinder or the piston from even the back end only, you have to:—(1) Disconnect the valve rods A R_2 and E R_2. (2) Disconnect the steam and the exhaust pipes S P_2 and E P_2. (3) To lift off the heavy cylinder cover C C, containing its steam and exhaust valves with their spindles, &c., without fouling the ends or faces of S P_2 and E P_2.

The port openings into the cylinder of the admission valves A V_1, A V_2 being horizontal, the pressure of the steam on the backs of these valves is not aided by their deadweights in keeping their working faces steam-tight to their ports in the same easy natural way, that they would do if their port openings led vertically downwards. Also, the valves will not be so well balanced, and there will be more tear and wear between the valves and their spindles than by the method referred to above.

Neither have springs for aiding steam-tightness of the valves been fitted to the valve-spindles, nor steam traps to the jackets of the Farcot-Corliss cylinders, as referred to above. Some engineers consider these springs superfluous; but now, no one can object to the use of the very best means of keeping the cylinder jackets as free of condensed steam as possible.



SECTIONAL VIEW OF A CORLISS CYLINDER WHICH COMPLIES WITH THE SEVERAL POINTS IN THE ABOVE CRITICISM.

Lecture XX.—Questions.

1. What advantages are claimed for the Corliss valve gear?

2. Give sketches of a Farcot-Corliss cylinder and its valve gear, showing the positions of the steam and exhaust valves, &c. Explain how the whole arrangement is fitted and works. Point out the weak points and make a design for a cylinder to fulfil the best up-to-date requirements of a Corliss cylinder, as mentioned at the beginning and the end of this lecture.

3. Mention the different types of Corliss valve gear. Illustrate by sketches the usual shape and construction of steam and exhaust valves for this gear.

4. Describe, with the aid of sketches, the original form of Corliss trip gear. Show the simultaneous and relative movements of the wrist-plate and steam valve levers.

5. Give a general description of the connections between and movements of eccentric, wrist-plate, valves and governor in the Corliss valve gear, together with a diagrammatic centre line view of the connections, and a complete index to the parts.

6. Illustrate and describe how the tripping of the valve is effected in the Farcot-Corliss valve gear.

7. Make a sketch of the Reynolds-Corliss double-eccentric valve gear, and explain its action.

8. Describe, with sketches, how the automatic lubrication of the various parts of a large steam engine which is not encased, is now usually performed. What is regarded as the best method of lubricating the cylinder of an electric light or power station engine? (B. of E., 1902, Adv. and H., Part i.)

9. Describe, with sketches, any form of sight-feed cylinder lubricator for use with the Seigrist system of automatic lubrication.

10. Sketch and describe how the system of forced lubrication is effected in a quick-revolution double-acting engine. Why are the working parts of such engines surrounded by a casing?

11. Sketch some common form of steam-engine governor, and show how it may be made to regulate the speed (1) by acting on the throttle-valve, (2) by varying the point of cut-off. Contrast the functions of a governor and a flywheel as speed regulators. (C. & G., 1902, O., Sec. C.)

12. Describe, with complete sketches, any form of steam engine governor with which you are familiar. (C. & G., 1901, O., Sec. C.)

13. Calculate the percentage saving in lbs of steam per B.H.P.-hour, and plot on squared paper the results obtained with the 300 B.H.P. triple-expansion Belliss-Morcom engine when using superheated steam of 160 lbs. pressure. (Use the data given in the table for this engine.)

14. Using the data given in the previously mentioned table, plot a curve, showing the change in efficiency for the dynamo-engine when run at 34 revolutions per minute for full, $\frac{3}{4}$, $\frac{1}{2}$, and $\frac{1}{4}$ loads.

15. Enumerate the several necessary precautions to be observed with superheaters and highly superheated steam.

16. Give a concise description, with sketches, of the Willans central valve engine, and note any outstanding features about this engine.

17. Calculate the percentage gain in steam for a 400-H.P. triple-expansion condensing Willans engine when using admission steam of

170 lbs. by gauge, but superheated from 0° to 180° F. Plot your results for gain in steam and for gain in B.T.U. (Refer to table of results in this lecture, and to the formula with curves in Lecture XVI.)

18. Answer only one of the following (*a*, *b*, *c*, or *d*):—* (*a*) Describe, with sketches, a Hartnell governor, and give the theory of its action. (*b*) Describe, with sketches, a loaded Watt governor, and give the theory of its action. State exactly in what way the load causes improvement in the action. (*c*) Describe any form of relay governor. (*d*) Describe how the governor acts in the case of the Corliss valve gear. (S. & A., 1897, Adv.)

19. Describe, with sketches, the construction and working of any crank-shaft governor which controls the advance and travel of a slide valve. (S. & A., 1898, Hons.)

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* Students should refer to the Author's Text-Book on *Applied Mechanics and Mechanical Engineering*, 3rd Edition (*et seq.*), Vol. II., for different kinds of governors.

LECTURE XXI.

CONTENTS.—Early History of Marine Engines up to 1815—Side Lever Engine—American Beam Engine—Steeple Engine—Double Cylinder Engine—Oscillating Engine with Valve Gear—Questions.

ALTHOUGH the successful commercial application of steam-power to the propulsion of ships was not effected until after Watt invented and perfected his double-acting engine, it will be interesting to briefly refer to a few of the more prominent attempts at steam navigation previous to and at the beginning of the present century.*

The earliest record that we can find of an actual attempt to propel a boat by a steam engine, is given in a correspondence between Papin and Leibnitz, wherein the former records having been present in 1698 at a trial of a boat driven by a Savery engine. The engine kept up a supply of water sufficient to work a water-wheel, which in turn drove the paddle-wheels. Papin, who was professor of mathematics at Marburg, had a vessel fitted with an engine of his own in 1707, wherein he employed the same device, viz., a pumping engine to force up water to turn a water-wheel attached to the propelling paddle-wheels. This vessel, however, before it had been put to regular use, was destroyed by a mob of boatmen who thought it would ruin their business. Papin himself narrowly escaped with his life and fled to England.

In 1736, Jonathan Hulls took out an English patent for a steam tug, in which the paddle-wheels were to be driven by a Newcomen's atmospheric engine, to which a system of ropes and grooved wheels, &c., was to be applied, so as to give a continuous rotary motion to the paddle-wheels placed at the stern of the tow-boat.

In 1783, the Marquis de Jouffroy, who was one of the earliest *savants* in France to recognise Watt's improvements, after several previous unsuccessful attempts, had a boat 150 feet long,

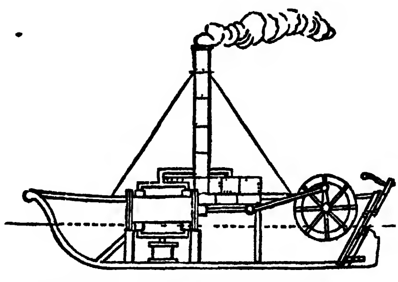
* For a complete history of the application of the steam engine to the propulsion of ships, the student is referred to Mr. Woodcroft's abridgements of patents, for marine propulsion, which will be found in most Engineers' and Philosophical Societies' Libraries, as well as to Prof. Thurston's *History of the Steam Engine.*

16 feet wide, fitted with a horizontal engine and paddle-wheels 14 feet diameter, 6 feet broad, and successfully tried it at Lyons, but owing to want of funds and discouragement from the French Government he did not put it to regular use.

In 1787, John Fitch made and tried a boat at Philadelphia, which was driven by side paddles worked by a steam engine, which attained a speed of 3 or 4 miles an hour; and in 1796 he experimented with a screw propelled boat at New York. This is the first actual trial of a screw propeller, although Daniel Bernouilli had in 1752 invented a form of screw propeller which he proposed to drive by a steam engine.

In 1788, Miller, Taylor & Symington, at Dalswinton, Dumfries-shire, Scotland, built and engined a small boat (25 feet long, 7 feet beam, with a double cylinder engine, the cylinders being only 4 inches diameter), which is reported to have attained a speed of 5 miles an hour. All these early attempts up to the beginning of the present century failed, chiefly on account of the imperfect means employed to transmit motion from the piston to the propeller. It was not until Watt's improved rotative engine began to be generally understood and appreciated that anything like practical success can be said to have been attained.

In 1801, Symington, encouraged by the previous partial success with Miller's boat, and availing himself of Watt's improvements, built and engined for Lord Dundas a small boat called the *Charlotte Dundas*, which plied as a tug-boat in 1802 on the Forth and Clyde Canal with complete success, and was only laid aside owing to the fear on the part of the canal directors that the wash from her propeller would injure the banks of their canal.



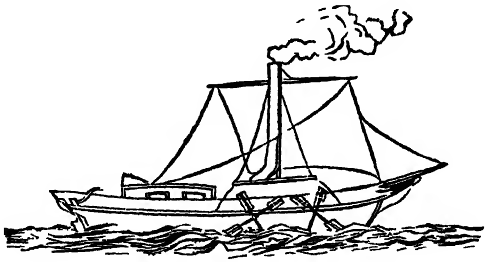
THE "CHARLOTTE DUNDAS," 1801.

As may be seen from the above figure, the vessel was fitted with a stern wheel, driven by a direct-acting horizontal engine with connecting-rod and crank. A condenser and air-pump were

fixed below the cylinder, while the boiler extended above the deck. Altogether the arrangement was most creditable, and she has justly been styled "the first practical steamboat."

In 1807, Robert Fulton, an American, had a steamer called the *Clermont* launched for him on the East River, New York, 133 feet long, 18 feet wide, and 9 feet deep, which he fitted with an engine having a cylinder 2 feet diameter, and 4 feet stroke, made for him by Boulton & Watt in England. This paddle boat made a trip to Albany, running the distance of 150 miles in 32 hours and returning in 30 without using the sails on either occasion. Old drawings, made by Fulton's own hand, of the *Clermont's* engine, are in the possession of Professor Thurston at the Stevens' Institute of Technology. The success of the *Clermont* on the trial trip was such, that Fulton soon afterwards advertised the vessel as a regular passenger boat between New York and Albany, and he has therefore the credit of first making steam navigation an every-day commercial success.

In 1812, Henry Bell constructed the *Comet* on the Clyde, a craft of 30 tons burden, 40 feet long, and 10½ feet beam, which ran for several years between Glasgow and Greenock as a regular passenger steamer.

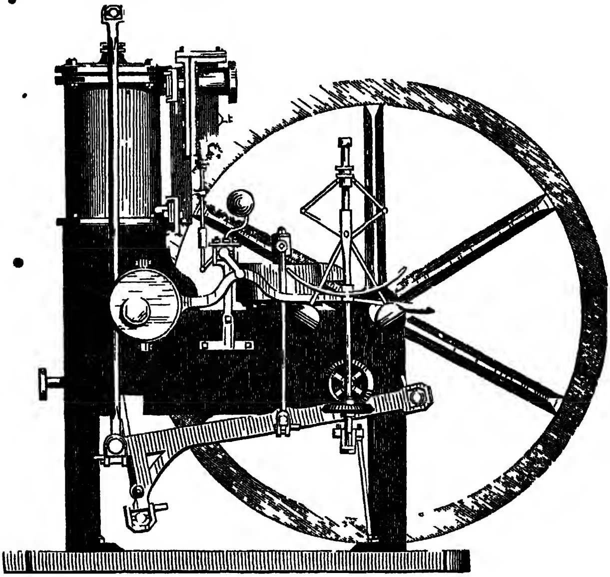


THE "COMET," 1812.

As may be seen from the above figure, there were two paddle-wheels on each side, driven by an engine rated at three horse power, of which the following diagram taken from Professor Rankine's *Steam and Steam Engine* gives an idea of its style and proportion.

This engine, as shown by the drawing, is what might be expected to have been used at the date of its construction for a small land engine, since it is fitted, not only with a fly-wheel, but also with a Watt's pendulum governor. It is a simple form of side-lever engine, where long return side rods from the

piston-rod crosshead engage with one end of a side lever, having a fulcrum or wyper shaft at its other end. With several important additions and improvements, such as jet or surface condensers, variable hand-regulated expansion gear, foot-trip and hand reversing gear, and omitting the fly-wheel and governor, this style of engine, termed a "grasshopper" engine, is to

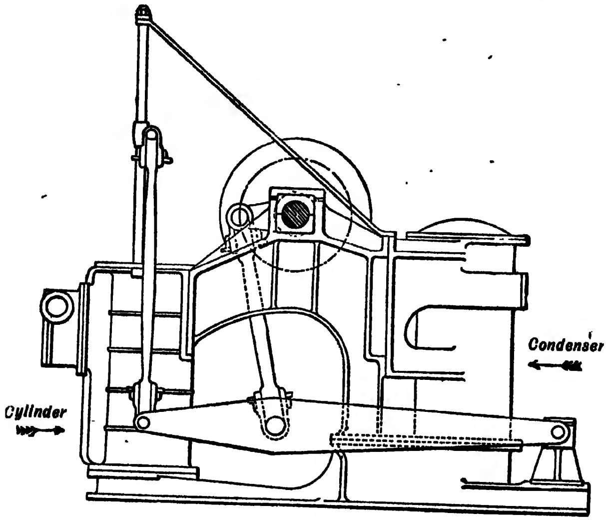


THE ENGINE OF THE "COMET."

be found at the present day doing good work in many tugs on the Thames, Clyde, Forth, and other ports. Several advantages are claimed for it, such as cheapness of construction, long stroke even in shallow water boats (where the cylinder is placed near the keel instead of on a raised platform as in the *Comet's* engine), less chance of sticking on the dead points than most other single cylinder forms of engines (owing to the position occupied by the crank shaft, the connecting-rod being placed between the cylinder side rods and the side-lever fulcrum), and also the fact, that it will work with less attention and in a greater state of disrepair than many other more finely adjusted forms of engines. The cylinder of the *Comet* is preserved as an interesting relic in the Glasgow Corporation Kelvingrove Museum.

From this date the advancement and success of steam navigation was very rapid, for we find that Bell soon built several

other steamboats. In 1814, there were 5 steamers in Great Britain (all Scotch) regularly at work in British waters; in 1820 there were 34, one-half in England, 14 in Scotland, and the rest



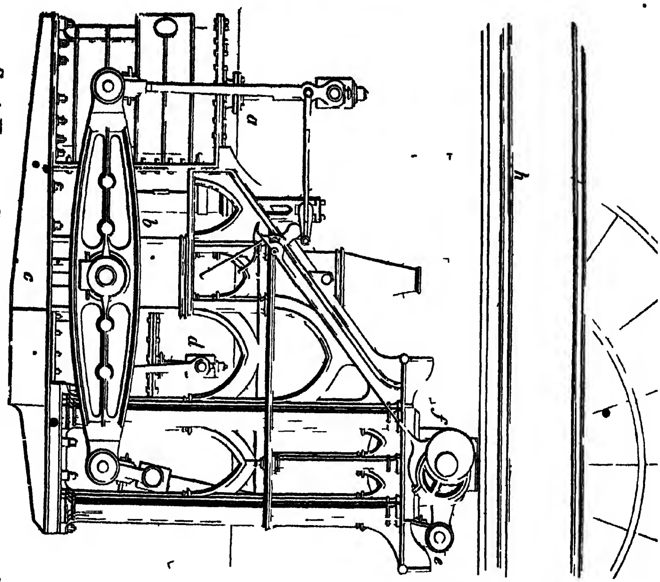
THE "GRASSHOPPER" ENGINE.

in Ireland. In 1840 there were 1,325 steam ships in Great Britain, of which 1,000 were English, and 250 Scotch In the year 1886, there were about 7,750* registered steamers belonging to all nationalities, and in 1884 on the Clyde alone, about 300 steamers of 290,000 tons burden were launched and fitted out.

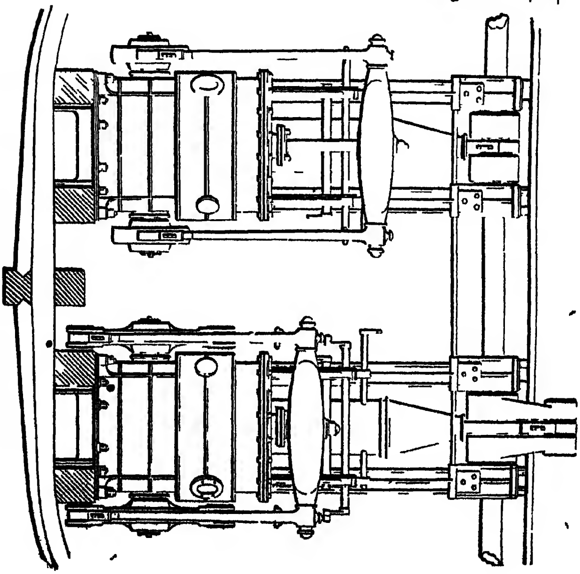
Side-Lever Engines.—In the earlier forms of engines for steamships, the propeller almost invariably used was the paddle-wheel, driven by what was known as the side-lever engine. This form of engine may be regarded as the marine counterpart of the land beam engine, so much in vogue in the early part of this century. This type of marine engine, although now entirely superseded in this country, was brought to great perfection by the Messrs. Napier of Glasgow, who fitted them to many of the most famous passenger ocean-going steamers prior to 1850.†

* 8,930 registered steamers belonging to Great Britain in 1888.

† The author remembers as late as 1866, being sent as an apprentice to assist at the repair of the engines of the old *City of London* (built and engined by Messrs. Robert Napier & Sons about 1840), which plied for



SIDE VIEW OF SIDE-LEVER MARINE ENGINES.



END VIEW OF SIDE-LEVER MARINE ENGINES.

The foregoing diagrams show the general arrangement of these engines:—The figure on the left hand is a side view of the port engine, while that on the right hand is an end view of the cylinders, &c., of both engines.

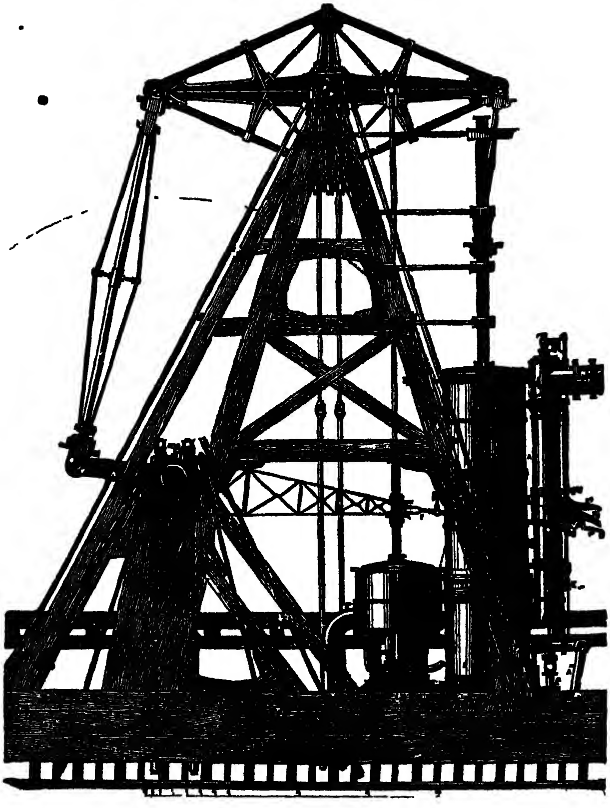
Each engine had a pair of side levers, *b*, fixed to one central rocking shaft, but on opposite sides of the steam cylinder, *a*. The piston-rod of the cylinder carried a crosshead like a large T (clearly seen in the end view), from which were suspended a pair of side rods, and those rods engaged with the after ends of the side levers. The other or forward ends of these side levers were connected to a single cross-tailed connecting-rod like an inverted T, thus ⊥, the upper end of which engaged the crank pin, *e*. The air-pump was also worked from the main side levers as shown at, *d*, while the jet condenser was situated between it and the cylinder. The eccentric with its counterpoise weight is seen at, *f*, and the paddle-wheel at, *h*. The whole engine rested on a heavy cast-iron girder sole plate, *c*. Such engines rarely used steam above 20 lbs. pressure on the square inch, and made about 18 revolutions per minute, or a piston speed of not more than 200 feet per minute, with a consumption of coal rarely less than 7 lbs. per indicated horse-power-hour; whereas now-a-days, a steam pressure of 150 lbs. with a piston speed of 600 feet, and a coal consumption of less than 2 lbs. are quite common. They were very heavy, occupied great space, and were often difficult to start, requiring in the larger boats sometimes two or more men at the starting wheel, for steam hydraulic starting gear, and balanced slide valves, had not been devised in those days, and only one eccentric was used, so that the slide valves had to be worked by hand until sufficient speed was attained to keep it in position for steaming either ahead or astern.

American Beam Engine.—This form of engine, which is peculiar to American river steamers, owes its characteristic design chiefly to Robert L. Stevens, the son of Colonel John Stevens, a contemporary and strong rival of Robert Fulton in shipbuilding and marine engineering at the beginning of this century. The "skeleton or walking beam" was first designed by Robert Stevens in 1822 for the *Hoboken*, and in 1827 he built the *North America*, one of the largest and most successful river steamers at that time. It attained the then extraordinary speed of between 15 and 16 miles an hour. This vessel had a pair of engines with single cylinders, each 44½ inches diameter, and 8 feet stroke,

many years between London and Aberdeen. The repair was necessitated by the breaking of one of the large side levers, *b*, a circumstance of not unfrequent occurrence with such engines. The diagrams on the last page give a good idea of her engines.

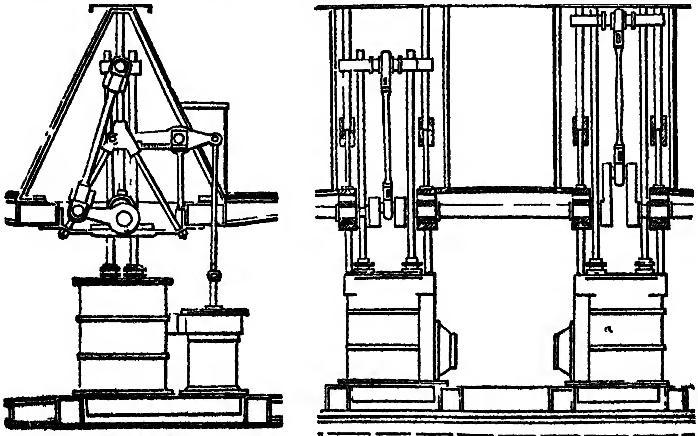
which made 24 revolutions per minute. In the vessel, he introduced for the first time, what is known in America as the "hog frame," a simple and efficient form of stiffening truss, for keeping long, light, and shallow vessels in shape when irregularly laden, and when steaming fast under the action of powerful engines

The following figure, taken by permission from Professor Thurston's *History of the Steam Engine*, clearly illustrates the common type of American beam engine, which has been even to the present day but slightly altered in general style since it was first introduced by Stevens, except that iron and steel take the place of wood in the "gallows frame," and a higher steam pressure, sometimes as high as 60 lbs. is now used :—



AMERICAN BEAM ENGINE.

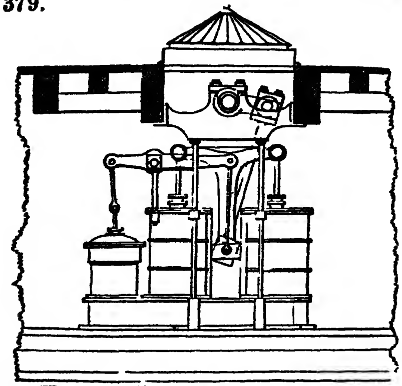
"This class of engine is usually adopted in vessels of great length, light draught, and high speed. But one cylinder is commonly used. The piston-rod crosshead is coupled to one end of the beam by means of a pair of links, and the motion of the opposite end of the beam is transmitted to the crank by a long connecting-rod. The beam has a cast-iron centre, surrounded by a wrought-iron strap of lozenge shape, in which are forged the bosses for the end centres, or for the pins to which the connecting-rod and the links are attached. The main centre of the beam is supported by a "gallows frame" of timber, so arranged as to receive all stresses longitudinally. The crank and crank shaft are of wrought-iron. The valve gear is usually of Stevens' form, the combined invention of Robert L. and Francis B. Stevens; the steam valves being of the disk or poppet variety, rising and falling vertically, and are four in number, two steam and two exhaust valves being placed at each end of the cylinder. The condenser is placed immediately underneath the steam cylinder. The air-pump is placed close beside the former, and worked by a rod attached to the beam. Steam vessels on the Hudson River have been driven by such engines at the rate of 20 miles an hour. This form of engine is remarkable for its smoothness of working, its economy and durability, its compactness, and the latitude which it permits in the change of shape of the long flexible vessels in which it is generally used without injury by "getting out of line."



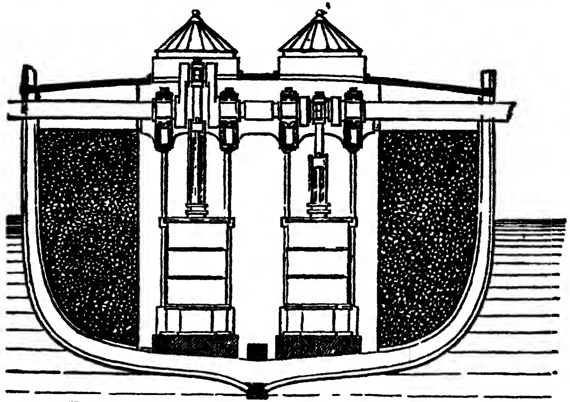
SIDE VIEW. END VIEW.

THE "STEEPLE" ENGINE.

It has, however, never found favour in this country, although several have of late been built on the Clyde by Messrs. A. & J. Inglis, of Pointhouse, for the River Plate South-American traffic. These boats had "gallows frames" of steel, but in their general features, they were similar to that shown in the illustration, p. 379.



SIDE VIEW.—"DOUBLE CYLINDER" ENGINE.



END VIEW.—"DOUBLE CYLINDER" ENGINE.

The Steeple Engine was one of the earliest forms of marine engine, and a great favourite on the Clyde for tug-boats and river steamers. It may still be seen in some of these older boats on the Clyde and elsewhere.

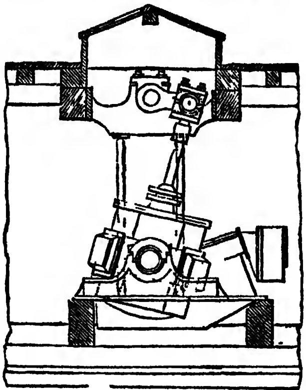
It possesses certain advantages over the side-lever engine, for it occupies less space, and is cheaper to make, having fewer working parts; but, on the other hand, the length of the stroke is limited by the depth of the ship, and considerable vibration takes place in the overhead guides, which rear high above the cylinder in a manner which renders them not so easily stayed.

Double Cylinder Engines.—One of the first of the direct acting type of marine engines, was that known

as Maudslay's (of London) double cylinder engine, a cross section and side view of which is shown in the preceding figure.

It consisted of two equal cylinders, placed side by side, of which there were usually two sets, as shown. In order to get sufficient length of connecting rod, the piston-rods of each pair of fore and aft cylinders were connected to one crosshead of T shape, the lower end of which dipped down between vertical guides placed betwixt the cylinders, and was there attached to the lower end of the main connecting-rod. The air-pumps were worked as shown from this same point by smaller connecting-rods and levers.

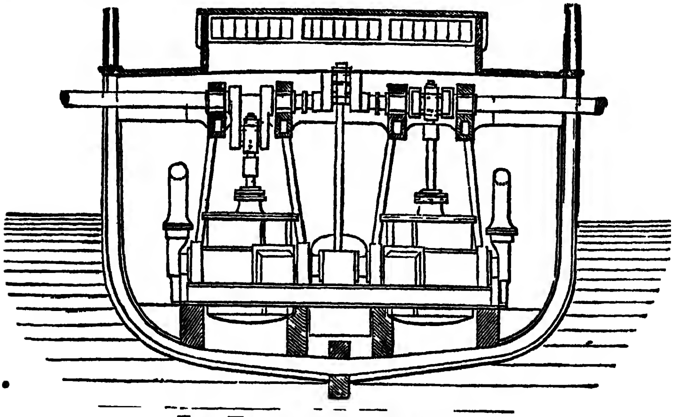
Oscillating Engines.—The oscillating engine was first used as a land engine, for we find that in 1785 Murdoch, the manager of Messrs. Boulton & Watts' engineering works at Soho, Birmingham, devised a simple oscillating engine. Trevithick is also reported to have suggested this form of engine, but it remained for the well-known firms of Messrs. John Penn & Son, of Greenwich, and Messrs. Maudslay & Field, of London, to perfect and adapt it specially to paddle steamers. The general arrangement is shown by the following figures, of which the left hand one is a side view, and the right hand one an end view, taken from Professor Rankine's *Steam and Steam Engine*:—



SIDE VIEW.—OSCILLATING ENGINE.

In these engines the chief feature is, that the connecting-rod is altogether dispensed with, the upper end of the piston-rod being supplied with an ordinary connecting-rod crank pin end, so as to work directly on the crank. The cylinder is usually placed vertically under the crank shaft, and is carried on two trunnions near the middle of its length, so that it may

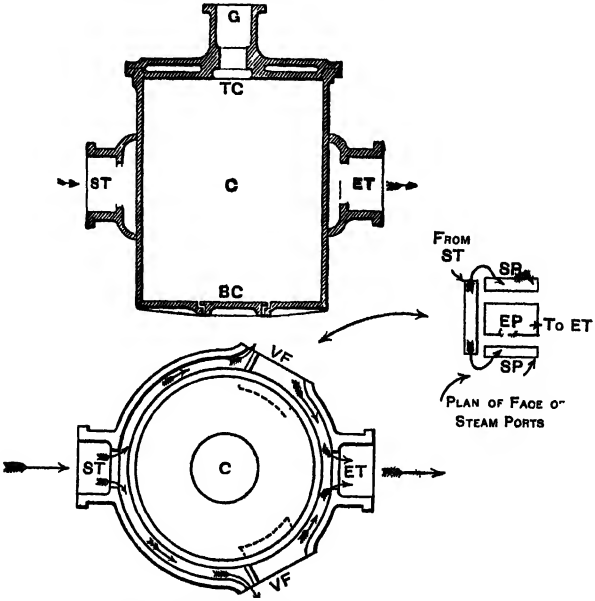
freely sway to and fro through a small arc, and thus permit the piston and piston-rod to follow the movements of the crank. From the following sectional elevation and plan (next page) of an oscillating marine engine cylinder, C, it will be seen, that the trunnions are hollow, the one next the skin of the ship being always the steam trunnion, S T, or that one connected directly to the steam pipe leading from the boiler, while the inner or central one is the exhaust trunnion, E T, connected directly to the condenser. Both are kept steam tight with a stuffing-box and gland. There are usually two valve chests bolted to the valve faces, V F, V F, placed on opposite sides of the cylinder, and at equal distances from the centre lines of the trunnions, so as to balance each other as they oscillate with the cylinder. A steam belt surrounding the cylinder connects the steam trunnion with the valve chests, and also the exhaust port of the valve casing with the exhaust trunnion; two diaphragms as shown, are cast in this belt to prevent communication between the entering and exhausting steam, except through the action of the slide valve. The top cover, T C, with its gland, G, and stuffing-box have to be made stronger and deeper than in ordinary engines with a connecting-rod and piston-rod crosshead guide, as they have to withstand the side stress of turning and stopping



END VIEW.—OSCILLATING ENGINE.

the momentum of the cylinder, and the piston-rod has also to be made larger for the same reason. A bottom cover, B C, is provided, for the purpose of facilitating the casting and boring out of the cylinder during manufacture, or getting in to clean out or to unscrew the piston-rod nut on the bottom of the piston. The

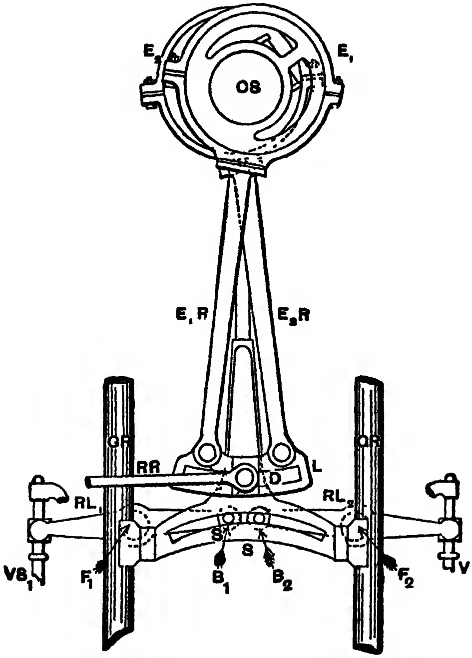
slide valves which are generally ordinary D slides, are worked by an ingenious arrangement from eccentrics on the crank shaft, as seen by the figure on the opposite page.



OSCILLATING CYLINDER.

Two eccentrics, E_1, E_2, generally cast and fitted on to the crank shaft, C S, in two parts, communicate an up and down motion (when one or other of them is in full or partial gear), through their eccentric rods, E_1 R, E_2 R, the link, L, and die-block, D, to a curved sector, S, moving between two vertical guides, G R. In this curved sector, S, are fitted two metal blocks, B_1, B_2, attached to the inner ends of two rocking levers, R L_1, R L_2, which transmit a simultaneous down and up motion to the cylinder valve spindles, V S_1, V S_2. The rocking levers with fulcra, F_1, F_2, are fixed to cylinder and curved round it to meet the valve spindles. The curvature of the sector is drawn with a radius from centre of trunnion. When it is required to reverse the engine, the reversing rod, R R, is moved to the right or to the left according as the engine is required to go ahead or astern, by the starting wheel, which is usually fixed on the platform on

a level with the top framing carrying the crank shaft. Either one or, if preferred, two air-pumps are worked at an angle from a central crank (shown in the end view), by means of a connect-



VALVE GEAR FOR OSCILLATING ENGINE.

C S	for	Crank Shaft.	S	for	Sector.
E_1, E_2	,,	Eccentrics.	G R	,,	Guide Rods.
E_1 R, E_2 R	,,	Eccentric Rods.	R L_1, R L_2	,,	Rocking Levers.
L	,,	,, Link.	F_1, F_2	,,	,, ,, fulcra.
R R	,,	Reversing Rod.	B_1, B_2	,,	Sector Blocks.
D	,,	Link Die-block.	V S_1, V S_2	,,	Valve Spindles.

ing-rod attached to a trunked plunger, while the condenser, if of the jet type, is placed between the cylinders, and if of the surface kind either before or behind them, but in the centre line of the ship.

Between 15 and 25 years ago, oscillating engines were by far the most popular kind of engines for fast passenger paddle-wheel steamers in this country. The oscillating engine was usually worked at a steam pressure of from 30 to 35 lbs. on the square inch, and produced most economical results at that pres-

sure, having sometimes as low a consumption as $2\frac{3}{4}$ lbs. of coal per indicated horse-power-hour. Now, however, engines of the compound type, with an early cut-off and expansion valve, can be made to work much more economically than this. A general feeling exists, that the trunnions of oscillating engines will not keep tight at very high pressures, such as 100 or more lbs., although some aver that there is no great practical difficulty in this respect. The oscillating form of engine does not seem to lend itself readily to compounding, and we do not hear of so many being ordered as formerly; direct-acting diagonal engines being seemingly preferred. However, where the economy of compound engines does not show to so great advantage, such as in the case of steamers making quick, rapid, short passages, with frequent stoppages, the oscillating engine is still a favourite, for it is the most compact and direct acting type of engine we have. It is easily started and stopped, the weight of the machinery is less than in most kinds, and is well down in the hull of the ship; moreover, the stresses are transmitted to, and readily taken up by the keelson, and the ship's frames.

Lecture XXI.—Questions.

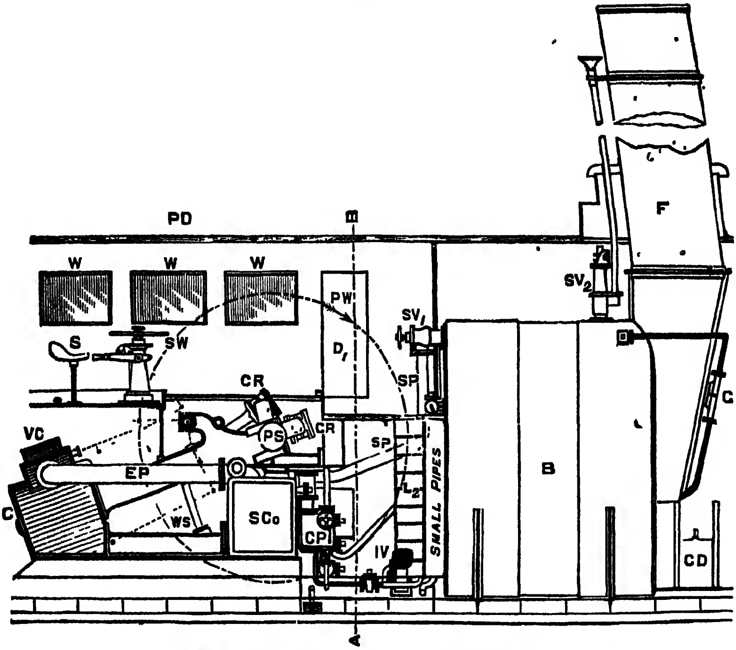
1. Referring to the early history of steam navigation previous to the beginning of this century, point out the chief causes of failure of early inventors.
2. Give an outline free-hand sketch of a "side-lever engine," with index of parts, using the first letters of the names of the parts, and state why this type of engine was given up, and when.
3. Give an outline sketch, with index of parts, of the "American beam engine," and state the advantages claimed for it, which will account for its retention, even to the present date, by the Americans What is the "hog frame," and of what use is it? Have you ever seen it or the beam marine engine applied in this country? If so, where?
4. Sketch in outline a "steeple engine" and a "double cylinder engine," with indices of the chief parts.
5. Sketch in outline a Grasshopper engine, and give an index of the chief parts. State the advantages claimed for this form of engine, and the kind of steamer for which it is best adapted. Have you ever seen an engine of this kind at work, and where?
6. Describe an eccentric and eccentric rod as fitted to marine engines, and show that they produce the same motion as a crank and connecting-rod. How is the eccentric connected with the slide valve in an oscillating engine? Give sketches and index of parts.
7. Describe the construction of the cylinder of an oscillating engine. Make a diagram showing how the slide valves are worked by the eccentrics as well as the steam and exhaust passages.
8. Describe, with sketches, the construction and arrangement of the cylinder, steam ports, and passages, together with the slide valve of a marine oscillating engine, and show the manner in which the valve gear is adapted to the oscillating cylinder.
9. Describe, with sketches, the construction of an oscillating engine, and the method of distributing the steam.

CONTINUATION OF LECTURE XXI.

CONTENTS.—Diagonal Direct-Acting Engines, with Joy's Valve Gear and Alley's Flexible Coupling, &c.—Paddle-Wheels—Radial Paddle-Wheel.—Feathering Paddle-Wheels.—Questions.

Diagonal Direct-Acting Engines.—This form of engine is very convenient, and is now the most popular for fast paddle-wheel river steamers of light draught. As will be seen from the illustrations which we give, it is neither more nor less than a horizontal engine set at such an angle, that the forward end is elevated to suit the necessary height of the paddle shaft, while the after end rests firmly on the ship's floor frames. It no doubt takes up a larger fore and aft space than the oscillating type, but, on the other hand, it occupies less space athwart-ship, and when the framing is carefully designed, using wrought-iron and steel wherever possible (instead of the older style of cast-iron framing), the weight per horse-power does not in all probability exceed that of its chief rival the oscillating engine. The weight is also better distributed along the keel of the ship, and the stresses set up by its action are chiefly in a fore and aft and downward direction, and therefore easily resisted by the natural structure of the vessel. Moreover, the chief working parts are in full view of the engineer while at the starting wheel, the engine is readily got at for adjustment and repairs, easily compounded, and all the most modern and efficient devices for quickly starting, stopping, and reversing, or for economising steam are easily applied to it. We illustrate one of the latest of these popular diagonal direct-acting engines, of which eight sets with their boilers, &c., were made last year (1885) by Messrs. Alley & Maclellan, Sentinel Works, Glasgow, for steamers now trading on some of the large rivers in India.

By studying the two figures, along with the index of parts, the student will be able to get a minute idea of the general arrangement of boilers and engines, but, in order that he may the better grasp the construction of the engines, we illustrate a perspective view of them as they lay in the workshop before being removed and fitted into the steamers. It will be observed that they are compound condensing engines, and are fitted with all the latest



General Longitudinal Arrangement.

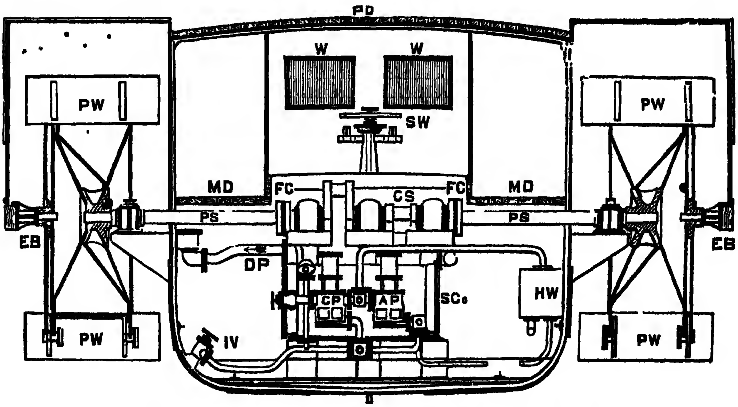
Diagonal Compound Engine and Boiler.

improvements, such as Joy's valve gear and Alley's flexible main shaft coupling. As these two improvements or modifications of them are fast coming into use, we shall illustrate and describe them.

Joy's Patent Valve Gear is a clever device, whereby the necessary motion of the slide valve, and the facility for reversing the engine, are effected by a series of links and connections between the connecting-rod and the valve spindle, thus replacing the ordinary double eccentrics and Stephenson's link-motion. The motion obtained by it is a very perfect one for a slide valve, for the travel of the valve is made quick on opening and on shutting off steam to the cylinder, and slow when the steam is expanding and exhausting. This is effected without any undue lead or com-

pression, or too early an exhaust. The space usually occupied by eccentrics on the crank shaft is saved, and thus the cylinders and the cranks, as well as the main bearings, can be brought much closer together.

At a joint, J, on the connecting-rod, C R, is attached a double link, L_1 L_1, about $\frac{1}{3}$ along this double link is attached a pair of

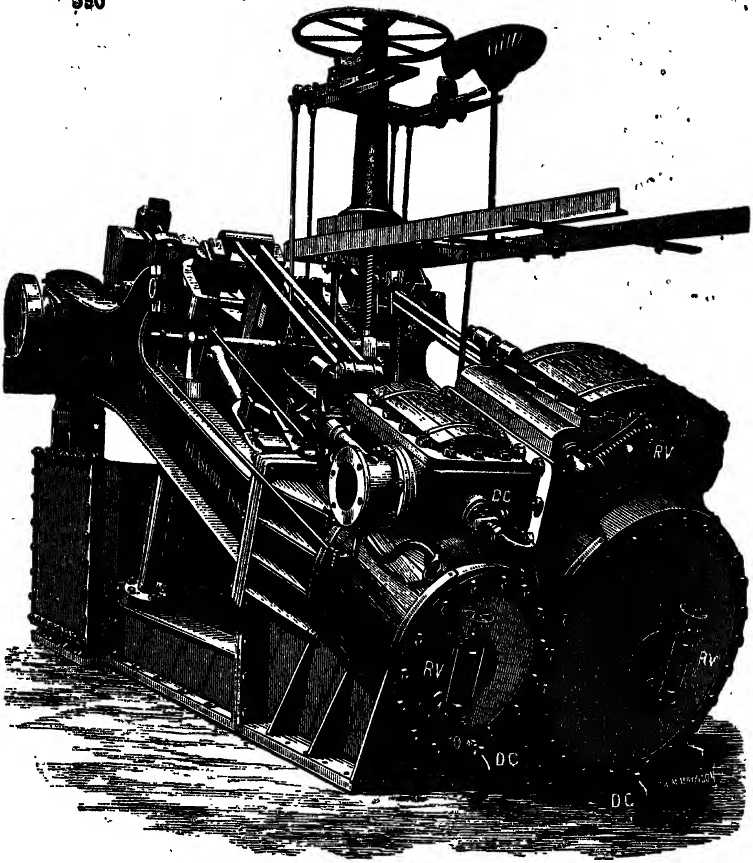


CROSS SECTION THROUGH LINE A.. B ON OTHER VIEW.

DIAGONAL COMPOUND ENGINES AND BOILER.

Index to Both Views.

P D	for Promenade deck.	E P	for Exhaust pipe.
M D	,, Main deck.	W S	,, Wrought-iron stanchion.
C D	,, Coal (bunker) door.	S Co	,, Surface condenser.
B	,, Boiler.	C P	,, Circulating pump.
F	,, Funnel.	I V	,, ,, inlet valve.
G	,, Gauge pipe to indicate height of water.	D P	,, ,, discharge pipe.
$S V_2$	,, Safety valves.	A P	,, Air-pump.
$S V_1$	,, Stop valve.	H W	,, Hot-well.
S P	,, Steam pipe.	C R, C R	,, Connecting-rods.
D_1,	,, Door to engine room.	C S	,, Crank shaft.
W, W,	,, Windows ,,	F C, F C	,, Flexible couplings.
L_2	,, Ladder ,,	P S, P S	,, Paddle shafts.
S	,, Seat for Engineer.	P W	,, Paddle-wheel.
S W	,, Starting wheel.	E B, E B	,, Outer eccentric bearings for feathering floats.
V C	,, Valve casing.		
C	,, Cylinder, low pressure.		

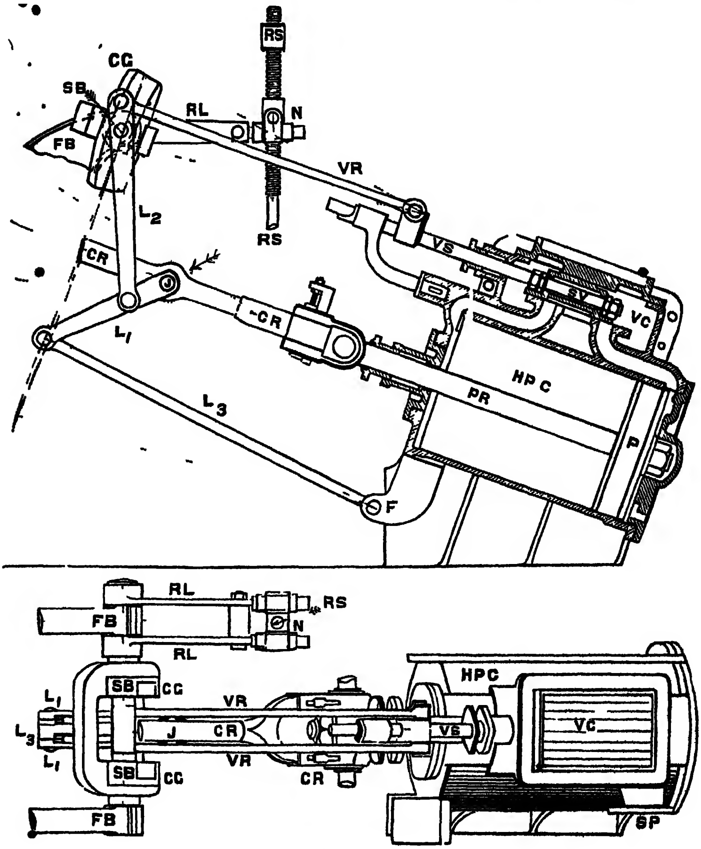


ALLEY AND MACLELLAN'S COMPOUND DIAGONAL ENGINES.

DIMENSIONS OF COMPOUND DIAGONAL ENGINES, &c.

STEAMER.		ONE BOILER (STEEL).		ENGINES.	
Builders' measurement	= 222 tons	Diameter . .	= 11 ft 6 in	Diameter H.P. cyl.	= 22 in.
Length . .	= 140 feet.	Length . .	= 9 ft. 6 in	" L P cyl.	= 37½ in.
Beam . .	= 18 feet	Thickness of shell	= $\frac{13}{16}$ in.	Length of Stroke	= 30 in.
Depth of hold .	= 7 ft. 6 in.	Furnaces . .	= Two.	No. of strokes per min.	= 44
Speed . . .	= 14½ kts.	Tubes . . .	= 192	Cut off in each cyl.	= 15 in.
Weight of Engines	= 22 tons.	" Diameter	= 3¼ in. O.D	Indicated Horse-Power	= 300
" Boilers in working trim	= 23 tons.	Grate surface .	= 33 sq. ft.	Diameter of Paddles	= 12 ft.
		Total heating surface	= 1298 sq. ft	Breadth . . .	= 6 ft.
		Pressure . .	= 100 lbs.	Immersion . .	= 2 ft. 3 in.

links, L_2, the upper ends of these being coupled first to sliding blocks, S B (working in curved guides, C G), and the extreme end to the valve connecting-rod, V R, which terminates at and is fixed to the valve spindle, V S, with its accompanying



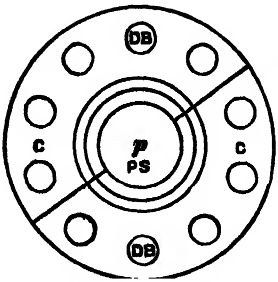
JOY'S PATENT VALVE GEAR.

slide valve, S V. The lower end of the double links, L_1 L_1, is connected to a radial rod or link, L_3, terminating in a fulcrum, F, fixed to a bracket on the high-pressure cylinder H P C,

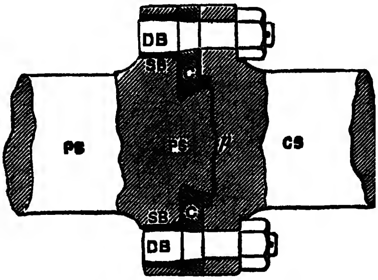
wherein is shown the piston, P, and piston-rod, P R. Precisely the same arrangement is carried out with respect to the low-pressure cylinder and slide valve. The double (curved) guide, C G, is free to move to the right or to the left in bearings carried by the main framing brackets, F B (see also the perspective and general views of the engine). The reversing or starting wheel, S W, is keyed at the upper end of the reversing screw, R S, which screw has a nut, N, fixed to the reversing levers, R L.

It will thus be seen, that the combined to-and-fro, up-and-down motion of the connecting-rod, is converted and transmitted to the valve spindle, in the form of a merely to-and-fro motion. The eccentricity of the connecting-rod's motion being duly corrected by the relative positions and lengths of the several links, L_1, L_2, L_3, &c., and that by elevating the nut, N, by the reversing screw, R S, the curved guides, C G, are turned backwards or from the cylinder, this action draws forwards the valve rod with valve spindle and slide valve, admitting steam behind the piston, and causing the engine to move forward, or in other words putting it in forward gear, while the depressing of the nut, N, effects precisely the opposite, and causes the engine to revolve backwards. All the joints are made very substantial and should be case-hardened, while their pins should be of steel to prevent wear and obviate rattling.

Alley's Patent Flexible Coupling.—The engines being described were fitted with this invention, which allows the paddle shaft to



FACE OF COUPLING.



LONGITUDINAL SECTION.

accommodate itself to the yielding of the paddle boxes and the hull of the vessel as it vibrates and changes form when working in a sea-way. It is equally applicable to the shafting of screw steamers. The hull of a vessel cannot be made absolutely rigid, and therefore it is wrong to make the shaft rigid. With

a rigid shaft and a flexible hull the result is an enormous amount of friction in the bearings, which consumes power, and often causes the bearings "to fire." Scarcely a month passes that we do not hear of some steamer breaking a main shaft, often to the danger of life and property, and this in many instances may be traced to a want of trueness in the line of the bearings, due to the vessel having warped from uneven stowage, or from having encountered heavy weather.

The coupling consists of a projection formed on the end of the paddle-shaft, P S, which is part of a ball, the centre of this projection being formed into a blunt point at, *p*. This point rests hard against the crank shaft, C S, and transmits any thrust along the line of shafting. The outside of the projection is clasped by the coupling ring, C, turned to fit the ball joint. This ring, C, is made in halves (as may be seen by the end view), and is secured to the crank shaft by means of the driving bolts, D B. The concave portion of this ring takes any pull that may come on the shaft along the line of shafting. The ends of the driving bolts, D B, project as shown into holes in the paddle shaft, P S, and thus act as drivers. These projecting pins are made $2\frac{1}{2}$ times the diameter of the bolts usually employed for main-shaft coupling flanges. These pins are slightly barrel-shaped in form, and made an easy fit for the holes in which they work. The holes are lined with hard steel bushes, S B, while the pins are case-hardened to prevent chafing and wearing away. It will be observed that there is a small space left clear between the paddle shaft flange and the coupling ring, C, to permit of perfect up and down or side play, or un-linement between the crank and paddle shafts for the reasons already mentioned.

Relief Valves.—On referring to the perspective view of the engines, it will be seen that relief valves, R V, are fitted not only to the front and back of the high- and low-pressure cylinders, but also to the back of the low-pressure slide valve casing, for the purpose of preventing damage to these parts through water gathering in them. These relief valves consist of ordinary mushroom valves, held down by strong spiral springs, and adjusted to any desired pressure by hand wheels and screws, as shown.

Drain Cocks, D C, are fitted to the back of the high-pressure slide valve casing, and to the bottoms of both cylinders. The pipes leading from them are all connected to the condenser. These cocks are opened before starting the engines, so as to clear away any water that may have resulted from condensation of steam, and also when the engines have to be stopped for any length of time.

Paddle-Wheels.—Having briefly described the different forms of engines used for driving paddle-wheels, we now naturally refer to the wheels themselves, leaving a description of the screw propeller until after we have noticed the styles of engines more particularly adapted to driving it.

The efficiency of the paddle-wheel falls off when the wheel is too deeply immersed, consequently for long voyages, where the draught of the vessel decreases as it proceeds, due to consumption of coal. &c., if the wheels are to be immersed to the proper depth at the end of the voyage, they must of necessity be too deep at the beginning. This variable immersion of paddle-wheels is the most serious objection to their use for long voyages. Also, in a heavy sea the *rolling* of the vessel, besides causing the engines to race, induces unequal straining of the machinery, since one wheel lifts out of the water, while the other sinks more deeply in it. Neither of these disadvantages is found in the screw propeller, for the screw is immersed considerably below the surface of the water, and since it is placed in the centre line of the ship, the rolling motion has no effect on it. The heaving of the ship in a fore and aft direction causes racing of the engines, but no unequal straining is set up. For short voyages, however, and where the draught is practically unchanged during the voyage, the paddle-wheel still holds its own with the screw, and for navigation in shallow rivers it is very valuable; the screw, in such a case, being quite unsuited, on account of its nearness to the bottom of the vessel. The vibration set up by the motion of paddle engines also is not so great as that from the fast-running engines necessary for the screw propeller.

Radial Paddle-Wheel.—This form of wheel is the simplest, strongest, least expensive, and least liable to derangement, but is also unfortunately the *least efficient.* It consists of radial arms, which are attached to a cast-iron boss at the centre, and are bound at their outer extremities by one or two wrought-iron rings. Flat boards are fixed rigidly to these radial arms, parallel to the axis of the wheel, and are known as "floats," and it is the thrust or push which these boards or floats exercise upon the water as the wheel rotates, which propels the vessel. The floats of a wheel of this kind, of necessity enter and leave the water in an oblique manner, and are only perpendicular to the surface of the water when they come immediately below the centre of the wheel. Therefore, since the pressure which a float produces is perpendicular to its surface (*i.e.*, perpendicular to the radius of the wheel), it is only when the floats are passing their lowest point, that the *whole* pressure they exert is utilised

in propelling the vessel; in all other positions, it is only the horizontal component of the pressure which exercises any propelling effect, and the greater the obliquity of action, the less is this horizontal component. A large proportion of the power spent in driving paddle-wheels of this form, is wasted in beating and churning the water with the floats, when these are in positions on either side of the vertical line through the centre. The deeper the immersion of the radial paddle-wheel, and the smaller its diameter for a given depth of immersion, the greater is the obliquity of action, and therefore the greater is the loss of efficiency.

Radial wheels are sometimes made in such a way that the floats can be quickly detached and fixed in positions nearer the centre of the wheel. This is advantageous, when from an increased load, the draught of the vessel becomes greater, thus causing greater immersion of the paddle-wheels, since then the diameter of the wheel is reduced, and thus by reducing the immersion of the floats, diminishes the loss from oblique action. This operation is known as *reefing* the paddle-wheels.

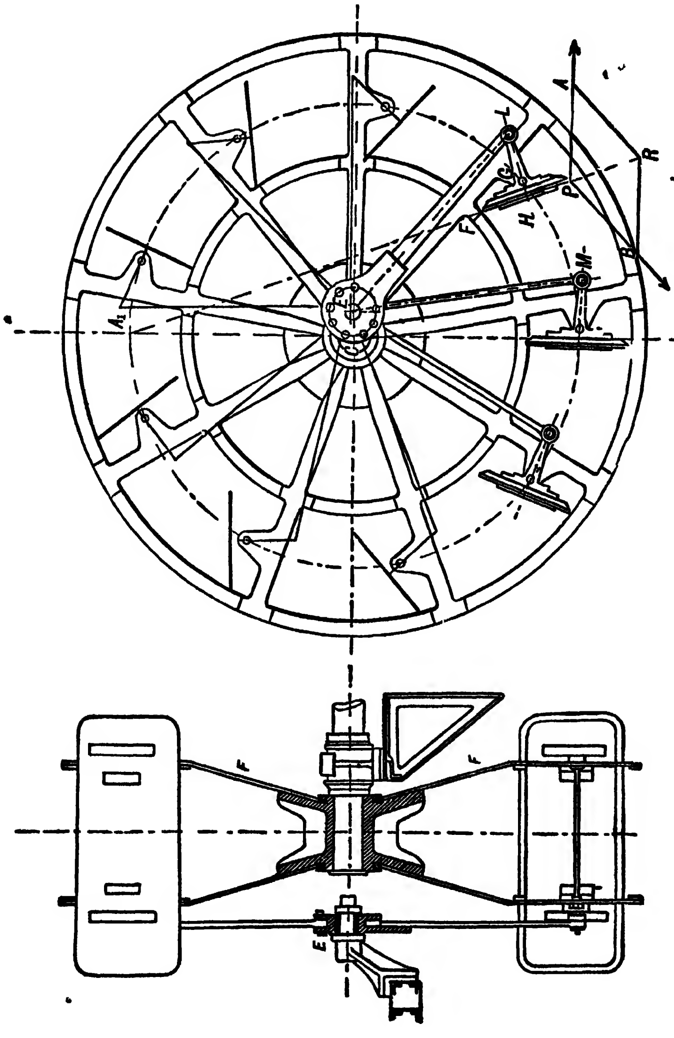
Radial wheels, double the diameter of feathering ones, are about equal in efficiency, but then the engine is quite twice as heavy. See American Engine.

Feathering Paddle-Wheels.—This form of paddle-wheel, as illustrated, is designed with the object of getting rid of the disadvantages which arise from the obliquity of action of the radial wheel.

The floats are not fixed rigidly to the radial arms, but are hinged to them, and are provided with levers, G L, so that they may be turned into any position. These levers are connected by rods to a boss or eccentric strap, E, which rotates on a central fixed pin on the sponson beam. This central pin is so placed in relation to the centre of the wheel, that the floats enter the water edgewise, and when in the water, are kept by the levers in a position nearly perpendicular to its surface. The radial arms of the paddle-wheel, F, are forged with small brackets on them at right angles to their lengths, so as to receive the round pins about which the floats hinge.

To find the position of the eccentric pin in order that the floats may enter the water edgewise, and pass through it nearly perpendicular to its surface, the following simple construction is necessary:—Suppose the lower edge, P, of a float to be just entering the water, then, in order that the float may enter edgewise, the resultant of its own velocity of rotation and the horizontal velocity of the vessel, must lie in the plane of the float. Let, P A, represent the velocity of the vessel, and, P B, that of

the wheel (drawn tangential to a circle which has, O, as centre, and passes through, P), then, completing the parallelogram, P R



PADDLE-WHEEL WITH FEATHERING FLOATS

is the resultant, and the plane of the float must contain P R. Produce, R P, to cut the vertical through the centre, O, at, A_1, and at right angles to the line, R A_1 (which is the line of the float); lay off, H G, equal to the distance from the face of the float to the centre of the hinge, and, G L, equal to the length of the lever. Now set off in outline another float immediately under the centre of the wheel, with its face perpendicular to the surface of the water, and having the end of its lever at M. With, M, and, L, as centres, and radius equal to, G O, describe arcs intersecting at, E, then, E, is the centre of the eccentric pin. Having thus determined the position of, E, the complete wheel may be drawn down and the proper pitch given to the floats. In actual practice the probable slip of the paddle-wheel has to be taken into account, and, therefore, a smaller circle than that with radius, C G, will be the rolling circle. The floats must, therefore, be so adjusted by moving the eccentric, E, that when entering and leaving the water they shall point to a position on the vertical centre line, considerably higher than the point, A_1, as shown by the right-hand figure. It is, however, advisable to make the floats enter the water a little flatter than the position so calculated for the assumed amount of slips, in order that the pressure of the water shall not come on the forward side of the floats. The speed of the paddle-wheel and of the ship should be carefully compared on the trial trips, and the eccentric shifted, if need be, until the best results are obtained. A considerable increase of speed of certain ships has been recorded by thus finding the most suitable place for the feathering eccentric.

The feathering paddle-wheel, although much more efficient than the radial wheel, is more liable to derangement, since any accident to the feathering apparatus would paralyse the action of the entire wheel. For this reason it did not find general favour when first introduced, but now it is almost universally adopted. It requires to be made specially strong, and all the pins and wearing parts should be cased with brass to prevent corrosion. The boss or eccentric, E, which carries all the rods for feathering the floats, runs loose on the fixed eccentric pin, and is turned round by one specially strong rod, known as the *driving* or *king rod*, shown at L E. The floats for large paddle-wheels are now frequently made of steel and curved slightly concave,)⟶ towards the direction of meeting the water when steaming ahead.

LECTURE XXI.—QUESTIONS (*Continued*).

1. Sketch and describe by an index of parts a side and an end view of the general arrangement of diagonal direct-acting engines, as fitted into a river passenger steamer, including the boiler.

2. Why are compound-diagonal direct-acting engines preferred to oscillating or other kinds of engines for shallow river paddle-wheel steamers?

3. Sketch and describe by an index of parts, side views and a plan of a compound direct-acting diagonal engine.

4. Sketch and describe by an index of parts, Joy's valve gear, pointing out its advantages and disadvantages as compared with eccentrics and link-motion.

5. Steamer main shafts often break, or their bearings give trouble by heating, account for this, and describe a plan or plans for alleviating this evil.

6. Sketch and describe a simple radial paddle-wheel. For what reasons has this form of wheel been abandoned?

7. Sketch and describe, by an index of parts, a modern feathering float paddle-wheel. What advantages has it over the older form of paddle-wheel?

8. Describe how you would design and construct the arms, floats, and feathering arrangements for a paddle-wheel.

9. Describe, with such sketches as you think necessary, some method of constructing a paddle wheel with feathering floats. Why has Buchanan's method, of causing the floats to dip into the water vertically, not been adopted in practice?

10. Describe Stephenson's link motion and Joy's radial valve gear, and explain precisely the effect of "screwing" or "notching" up in each case. Sketch the indicator cards in order to illustrate your remarks. (C. & G., 1902, H., Sec. B.)

CONTINUATION OF LECTURE XXI.

CONTENTS.—Early Invention of the Screw Propeller—Geared Engines—Penn's Trunk Engine—Maudslay's Return Connecting-rod Engine—Horizontal Direct-Acting Engine—Vertical Direct-Acting Engines—Questions.

Early Invention of the Screw Propeller.—As we remarked before, when reviewing the early history of the marine engine prior to the beginning of this century, Daniel Bernouilli invented in 1752 a screw propeller which he proposed to drive by a steam engine, and John Fitch experimented with a little screw steamboat on the "Collect" Pond, New York, in 1796. In 1804, Colonel John Stevens of Hoboken, America, completed a steamboat 68 feet long, and 14 feet beam, which he fitted with a water tubular boiler, and a direct-acting high-pressure condensing engine, having a 10-inch cylinder of 2 feet stroke, driving a screw with 4 blades, and of a form which even at the present day appears quite good.*

A model of his boiler, engine, and screw, is preserved in the Mechanical Engineering Lecture Room, at the Stevens' Institute of Technology. In 1805, Stevens built another boat, introducing twin screws. Several other engineers proposed, and some of them tried, screw propulsion, but it was not brought into general use until John Ericsson, a Swedish engineer residing in England, and E. P. Smith, an English farmer, perfected and pushed its introduction in Great Britain, and in America, in 1836-37.

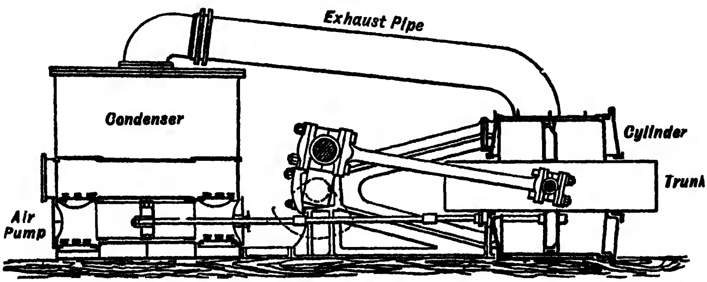
Geared Engines.—Within a few years from this date, the style and speed of steamship engines became entirely altered from what had been used in connection with the paddle-wheel; yet, engineers naturally tried at first to adapt the then existing forms of paddle-wheel engines to drive the screw. The screw has, however, to be run at many more revolutions per minute than the paddle-wheel, and since engineers in those days regarded anything over 200 feet per minute of piston speed as dangerous, or likely to derange their machinery, they preferred to get up the necessary speed by gearing. Thus, beam, side-lever, oscillating, and some of the various other forms of engines already mentioned in Lecture XX., were made to do duty in driving the screw propeller by means of stepped cog-wheels. Shortly after the successful commercial introduction of the screw as a propeller for

* See Prof. Thurston's *History of the Steam Engine.*

merchant ships, the Admiralty were induced to build two ships of the same model and size, viz., the *Rattler* and the *Alecto*, fitted with engines of the same power, but the former was provided with a screw, and the latter with paddle-wheels. A series of competitive trials were made with these two vessels, and the great success of the *Rattler* so satisfied the Admiralty and all engineers of the advantages possessed by the screw, that it very soon came to be generally adopted for ocean-going steamers. By gradual steps and improvements in the arrangement, and construction of the machinery, direct-acting fast-speed engines were adopted, until nowadays a piston speed of 700 feet per minute is not uncommon.

We now propose to briefly notice a few of the most successful styles of screw-driving engines before explaining the screw itself, and in a future lecture we shall describe in full detail a set of compound inverted cylinder vertical engines. In the navy, where the machinery has to be placed below the water line, the three principal types of horizontal engines that have in turn found favour with the Admiralty, are—(1) Trunk, (2) Return Connecting-rod, and (3) Horizontal direct-acting. In the merchant service, the vertical inverted-cylinder direct-acting engine has been generally adopted for the last thirty-five years.

Penn's Trunk Engine.—The difficulty of obtaining a sufficiently long stroke from the direct-acting horizontal engine in the case of a man-of-war, where the engines had to be placed as near the keel of the ship as possible, was solved by Mr. John Penn of



Greenwich. He hinged the connecting-rod direct to the centre of the piston by means of a gudgeon, surrounded by a brass cylindrical case or trunk, concentric with the steam cylinder, as seen in the following figure. This trunk was fixed to the piston, and protruded from each end of the cylinder through stuffing boxes, thereby not only giving additional support to the piston, but also permitting access for oiling the gudgeon and

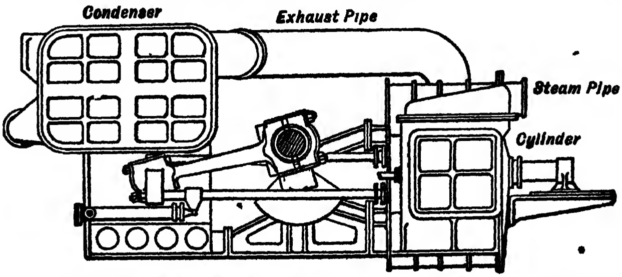
connecting-rod end, and preserving an equal area to the pressure of the steam on both sides of the piston.

Seaton, in his *Manual of Marine Engineering*, says—"This engine is the lightest and most compact of all the forms of marine screw engines, when constructed of the same materials; and for large sizes with the *lower steam pressures*, has been unsurpassed by any other type of engine. The length of stroke is considerably more than that of the ordinary direct-acting engine, and the connecting-rod much longer than that of any other form, being from two and a half to three times the length of the stroke; the weight of the piston is taken by the trunks in a great measure, and there are no piston-rod guides. But with the increase of pressure the defects of this form become more apparent, and lie with the very part that distinguishes it—viz., the trunk.

"The friction of the large stuffing-boxes is very great; in fact, may be so great by unduly tightening the glands as to stop the engine. The loss of heat from the large surface of the trunks being alternately exposed to steam and to the atmosphere, is very great, as is also that from their inner surfaces. The gudgeon brasses are exposed to a very high temperature and liable to become heated, and when heated cannot easily be cooled, as from their position they are not readily adjusted."

Penn arranged his engine so that the direction of motion of its crank when going ahead caused the thrust of the connecting-rod to be upward, and thus, as far as possible, to relieve the bottom of the cylinder from the tear and wear due to the weight of the piston. Some of the largest and most powerful ships in the British Navy have been engined with this Trunk form, such as—H.M.S. *Neptune*, 9000 I.H.P., H.M.S. *Sultan*, *Hercules*, *Minotaur*, *Northumberland*, *Warrior*, *Black Prince*, *Devastation*, &c.*

Maudslay's Return Connecting-Rod Engine.—Another modification of the horizontal engine, or rather of the old steeple



* A number of these ships have been re engined with more modern types.

form, is that known as the return connecting-rod, by which the same object is attained as in the last type, viz., a sufficiently long stroke and connecting-rod in the narrow cramped space of the hold of a vessel. The general arrangement will be at once understood from the preceding figure.

By the above design, the cylinder may be got close up to the turning range of the crank pin and connecting-rod head, that a longer stroke is obtainable than by any other plan of horizontal engine arrangements. The difficulty in small engines of getting in a small high-pressure cylinder alongside of the larger low-pressure one (with the object of compounding the engines), owing to the necessity for two piston-rods from each cylinder clearing the crank shaft, is overcome in most instances by placing the high-pressure cylinders immediately behind the low-pressure ones—tandem fashion—with one piston-rod only, protruding from behind each of the low-pressure cylinders, and attached to the high-pressure piston. The chief objections urged against this form of engine are—(1) The double piston-rods from the front ends of both cylinders; this entails double the number of stuffing-boxes, and the keeping of the crank shaft bearings from being close to the crank arms. (2) The eccentric-rods are also of necessity very short, unless placed, as is sometimes done, on the same side as the connecting-rod. The first engines of H.M.S. *Monarch* and *Raleigh* were of this type, and had four piston-rods to each cylinder.

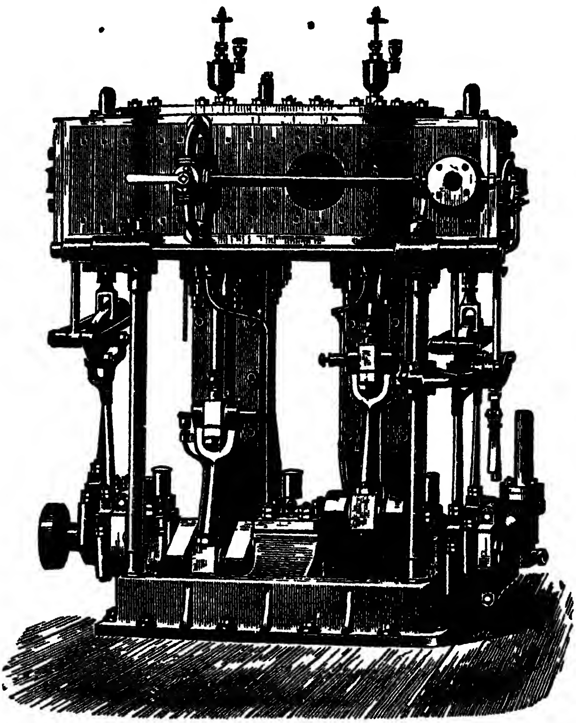
Horizontal Direct-Acting Engine.—This form of engine having its connecting-rod directly between the piston-rod crosshead (on the cylinder side) and the crank is certainly the simplest and most convenient type for a gun-boat or large naval vessel, where sufficient room can be obtained. Most of the late Admiralty orders are being constructed on this plan, *e.g.*, the *Australia* and *Galatea* by Messrs. Robert Napier & Sons, and the six ships of the *Scout* and *Archer* class* by Messrs. J. & G. Thomson of Clydebank.

They have the same essential parts, and work on the same principle as the compound inverted-cylinder engines which we shall describe in Lecture XXII.; and we only omit explaining and illustrating their distinctive features and details from want of time, space, and diagrams at our disposal.

Vertical Direct-Acting Engines.—The simplest form of marine engine used for small tug-boats and for steam launches at the present time, is that of the compound inverted-cylinder non-condensing type. The following illustration shows the general

* For perspective views and a description of the *Scout's* engines, see *The Engineer*, December 18, 1885.

arrangement of one of a pair of these small engines manufactured by Messrs. Alexr Shanks & Son, of Arbroath; from which it will be seen, that the high and low-pressure cylinders are supported at the back upon two cast-iron columns, and at the front by two wrought-iron stanchions. All four supports are fixed to a strong cast-iron sole-plate, which is bolted to the ship's floors. The back columns form the guides for the cross-heads of the piston-rods. The valve casings are placed on the



SHANK'S COMPOUND NON CONDENSING ENGINE.

fore and aft ends of the cylinders, which admits of the slide valves being readily inspected and adjusted. The slide valves are worked and reversed by the ordinary double eccentrics with link-motion. A boiler feed-pump is worked from one end of the

crank shaft, and a bilge-pump from the other end, both being driven by eccentrics, &c., as shown. The whole of the outside moving parts are easily got at for oiling and for adjustment, and all wearing parts are arranged so that the wear and tear may be readily taken up It will be observed that the upper ends of both connecting-rods are provided with projecting pins on their inner ends. This is for the purpose of working air and circulating pumps by means of levers, should it be found desirable to work the engines as condensing engines, in which case, a surface-condenser is placed separate from them in some convenient corner of the engine-room. The speed of these engines varies from 230 to 300 revolutions per minute.

LECTURE XXI —QUESTIONS (*Continued*).

1. On the introduction of the screw as a ship's propeller geared engines were at first adopted, why? What advantages have direct acting over geared engines?

2 Sketch a section, through the cylinder, air pump, and condenser of Penn's trunk engine Describe generally the arrangement of the engine, and show the connection of the piston with the screw shaft Why is this style of engine being discontinued in the Navy?

3 Describe, with a sectional sketch, Maudslay's return connecting rod engine, and point out its advantages, and disadvantages. In what class of ships are Maudslay's and Penn's horizontal engines used, and why?

4 What style and arrangement of engine is now being chiefly ordered by our Admiralty, and why?

5. How would you arrange the cylinders for compounding a pair of simple condensing Maudslay's return connecting rod engines?

6 Give a general outline freehand sketch, with concise description, of a pair of inverted cylinder compound non condensing engines For what classes of ships is this style of engine suitable, and why?

7 Sketch and describe an escape valve as fitted to the cylinder of a marine engine. Why is such a valve required, and where is it placed?